LOCOMOTIVE PERFORMANCE

THE RESULT OF A SERIES OF RESEARCHES CONDUCTED BY THE ENGINEERING LABORATORY OF PURDUE UNIVERSITY

BY

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PREFACE.

For a number of years the Engineering Laboratory of Purdue University has concerned itself with problems relating to the performance of locomotives, and the results of its researches have from time to time appeared in the proceedings of various scientific and technical societies. This process of publication has extended over a period of fourteen years, and has run through many different channels, with the result that the record now exists in widely scattered parts, which are often difficult of access and, therefore, of limited usefulness.

The purpose of this volume is to combine the most important of these results with other material not before published, and thus make a permanent and accessible record of the work of the laboratory. If it should appear that the pages have been burdened with too great an array of detail, it should be rememhered that many prefer such a minute presentation of facts as will enable them to work out conclusions for themselves. Primarily, the volume is designed as a record rather than a text, though it is hoped that it will prove of interest and value to any who wish to increase their acquaintance with the action of steam locomotives.

As a whole, the researches are the outcome of many influences. The trustees and president of the University have supplied means, professors and instructors have assisted in working out the problems of the laboratory, students have given their aid as observers, and in addition to these, skilled assistants have had a part in checking and arranging data.

Much work and many plans necessarily preceded the actual work of the laboratory, and whatever may have been accomplished in the development of a system for testing locomotives is due, in the first instance, to the part which was taken by the late President Smart. It required some courage for the president of a university not rich in funds to respond to the suggestion of a department, and,

in the absence of favorable advice, and even in the face of some adverse criticism, to give his approval and support to an untried process involving the expenditure of a considerable amount of money; but President Smart carefully considered every favorable plea and weighed every objection, and when he finally decided to enter upon a line of work so novel as that of testing locomotives in a laboratory, he took a bold but nevertheless a well-considered and intelligent step. He afterward imparted so much of his spirit of enthusiasm to the Board of Trustees as led them out of the limited resources at their disposal to supply means for executing his plan; and later still, when difficulties appeared in the development of mechanical matters, he was ever patient with delays, helpful in advice, and unfailing in his support.

Another whose help counted for much was the late A. J. Pitkin while General Superintendent of the Schenectady Locomotive Works. In order to make the undertaking a success, it was necessary that funds at the disposal of the University should be supplemented by outside aid, and it was through the friendly influence of Mr. Pitkin that arrangements were finally made by which the Schenectady Locomotive Works agreed to supply a suitable locomotive in return for the small sum which was available. Without this coöperation the establishment of the testing-plant would have been greatly delayed, or it would have proceeded under conditions far less favorable than those which afterward existed. Among others who early lent encouragement to the work of the testing-plant should be mentioned Professor James E. Denton, Mr. William Forsyth, and the late Mr. David L. Barnes. These mechanical engineers were among the first visitors who came to inspect the initial testing-plant, and their subsequent interest and frequent commendations went far to gain for the laboratory a degree of recognition which, in many incidental ways, has ever since been helpful. It is impossible to acknowledge properly the assistance of every one concerned in the actual operation of the laboratory, but mention should be made of Mr. Richard A. Smart, who for a considerable time was in immediate charge of the testing-plant, and of Messrs. Daniel Royse and Robert S. Miller, who each for a period of one year checked the numerical work of students.

W. F. M. G.

PURDUE UNIVERSITY,
LAFAYETTE, INDIANA, March, 1906.

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LOCOMOTIVE PERFORMANCE.

I. LOCOMOTIVE TESTING.

CHAPTER I.

THE DEVELOPMENT OF THE PURDUE TESTING-PLANT.

1. The Growth of Engineering Laboratories at Purdue.—Purdue University was opened as a school of Science, Agriculture, and Mechanic Arts in 1874. Five years later instruction was given students in shop-practice, and in 1882 a regular four-years course in Mechanical Engineering was established. From this time there was a gradual increase in the amount of apparatus available for the use of engineering students, but it was not until ten years after the establishment of the courses in shop-work that the development of an engineering laboratory was entered upon.

In the spring of 1890 the policy with reference to the equipment of such a laboratory found expression in an order given for a cross-compound Corliss engine which during the summer of that year was established as a complete testing-plant with condenser, air-pump, weighing-tanks, and all other accessory apparatus needful for experimental work. A year later, in 1891, a considerable sum of money became available for use in the extension of laboratory facilities, and, with a view to future development, plans for an extensive building were adopted. Two important results followed: one consisting in the erection of a portion of the building which had been planned, and the other in the establishment of a locomotive testing-plant as a part of its equipment.

2. Considerations Leading to a Locomotive Testing-plant.—The locomotive testing-plant was the outgrowth of natural condi-

tions. The Trustees and the President of the University had already shown their interest in the development of laboratories for engineering students, and it was easy for them to foresee the great advantage to be derived from a study of locomotive performance under conditions as favorable to the work of testing as those which had been so often declared necessary in connection with stationary engines. It was known that under conditions of service locomotives were tested with difficulty, results being far less satisfactory than similar results obtained from stationary engines under service conditions. If, in stationary practice, progress had been made by the establishment of experimental plants, how much more might be achieved by the installation of experimental locomotives!

These and similar arguments made it clear that the process of building up the equipment of an extensive engineering laboratory might easily and logically be made to involve the erection of a locomotive testing-plant, the value of which both as a means to the instruction of students and for the purposes of research could not be doubted.

The general outlines of the plant having been defined, its design and establishment became largely a matter of routine. The locomotive, stripped of the enchantment which may perhaps attend it on the road, became simply a steam-engine and a steam-boiler. Its action was not different in principle from that of other engines and boilers already included in the plans of the laboratory, and hence great experience in the management of locomotives, as such, was not necessary. In fact, the Purdue staff, which was instrumental in establishing the plant, had no member who had been trained in the motive-power department of a railroad.

3. **Arrival of the Locomotive.**—In September, 1891, the locomotive which had been named for its builders, "Schenectady," arrived upon a switch of the Lake Erie & Western Railway about one mile distant from the laboratory, and at a point having approximately the same elevation with the University grounds. There was no track to the laboratory. The surface of the ground between it and the laboratory which was to be traversed was slightly rolling, and from a considerable portion of the intervening territory a wheat-crop had been taken eight or ten weeks previous to the arrival of the engine.

The delivery of the engine upon the switch aroused great enthusiasm on the part of the students of Purdue, and out of respect for

the interest shown a holiday was declared, and a call was made for volunteers to assist in receiving the engine and in starting it on its overland journey. It was rather late in the morning before the work began. The privilege of breaking a joint in the track, to assist in rolling the engine out upon the temporary skids which had been prepared, having been denied, it was found necessary to block and carry the flange of the wheels over the top of the switch-rails, a task which involves some difficulty even when managed by experienced men with plenty of tools, whereas in the case described both experience and tools were lacking. But many willing hands, aided by a single team of horses, made light work of what might otherwise have been a laborious undertaking, and when night arrived "Schenectady"

FIG. 1.—The Purdue Locomotive, Schenectady No. 1.

had been pulled across the right-of-way ditch and well out toward the middle of the first field.

By the close of the first day the measure of the task had been taken and time was therefore given to a reconstruction of equipment. Three sections of track were made, each of a rail's length. built in the form of skids and capped by 56-pound rails. The foundation of each consisted of two 5×12 yellow-pine pieces, laid flatwise, across which 2×12 pieces were spaced as ties, the rail-spikes passing through the ties and into the foundation beneath. The reorganized force included three pairs of horses with drivers and two or three men to handle blocking. One pair of horses was employed to give forward movement to the engine; a second to draw the skids one after another from rear to front; and the third to pull the heel of the advancing skid

into line with the one previously placed. The men soon became so skilled in their respective parts that where the ground was smooth the engine could be kept in constant motion for considerable distances, one skid being drawn from rear to front and placed in position while the engine was passing over the other two.

Even with the new apparatus it was found impossible to make the locomotive follow the skids if laid on a curve, and wherever a change of direction was necessary it was made by laying cross-blocking under the skids, upon which one end of the skid bearing the engine was slipped bodily. The course involved four turns, each somewhat less than a right angle, and the whole distance

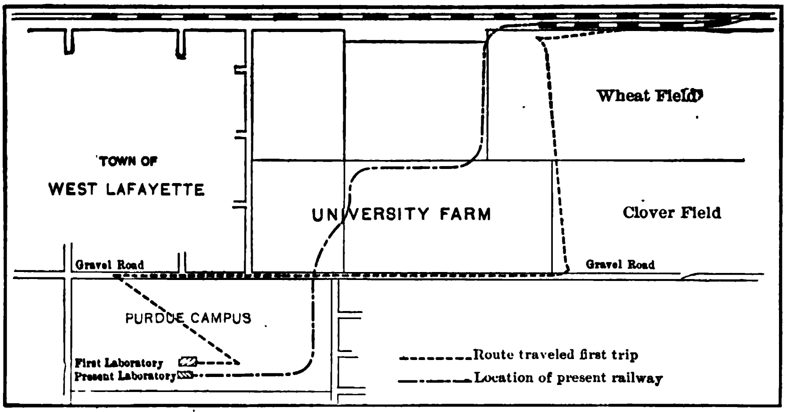

FIG. 2. -The Course followed from Track to Laboratory.

traversed by the engine was in the neighborhood of 1½ miles. On the eighth working day after the start, the engine arrived at the laboratory without accident and without having once touched the ground. *

4. The First Testing-plant.—It has been already noted that before the locomotive was ordered, lines had been laid down for an extensive engineering laboratory, and the construction of a portion of this had been entered upon early in the spring of 1891. The building thus begun constituted a definite portion of the larger struc-

* The success attending the transportation was chiefly due to the skill and energy of Mr. Robert Lackey, who had immediate charge of the operations after the first day's work. Mr. Lackey at the time was a student-assistant in the laboratory. By his good judgment an undertaking which, if less perfectly managed, might have been an expensive one, cost but a trifling sum.

ture of which it was eventually to form a part. A view of this portion of the building is shown by Fig. 3, and the location of the testing-plant within it by the floor-plan, Fig. 4.

FIG. 3.—The Engineering Laboratory, 1891.

Between the time of ordering the locomotive and its delivery, the details of the mounting mechanism were designed and put in place, so that when, in September, 1891, the locomotive arrived, the

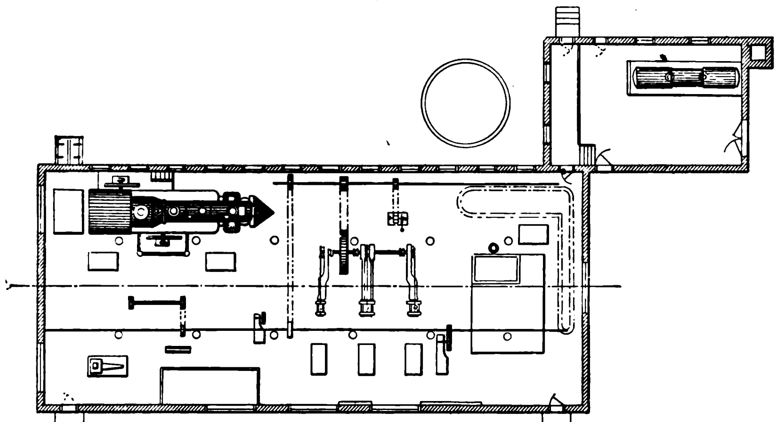

FIG. 4.—Plan of the Engineering Laboratory, 1891.

plant was practically ready for its reception. The following description of this first locomotive testing-plant is based upon a paper presented before the American Society of Mechanical Engineers.*

* "An Experimental Locomotive." Proceedings of the American Society of Mechanical Engineers, 1892.

The plan of mounting, in its inception, involved (1) supporting wheels carried by axles running in fixed bearings, to receive the locomotive drivers and to turn with them; (2) brakes which would have sufficient capacity to absorb continuously the maximum power of the locomotive, and which should be mounted on the axles of the supporting wheels; and (3) a traction dynamometer of such form as would serve to indicate the horizontal moving force and at the same time allow but a slight horizontal motion of the engine on the supporting wheels. It was believed that a locomotive thus mounted could be run either ahead or aback under any desired load and at any speed; that while thus run, its performance could be determined with a degree of accuracy and completeness far exceeding that which it is possible to secure under ordinary conditions of the road; and that the whole apparatus would be extremely valuable to students in steam-engineering. It was not assumed that every condition of the track would be perfectly met, but it was expected that the results obtained would prove valuable in extending a knowledge of locomotive performance.

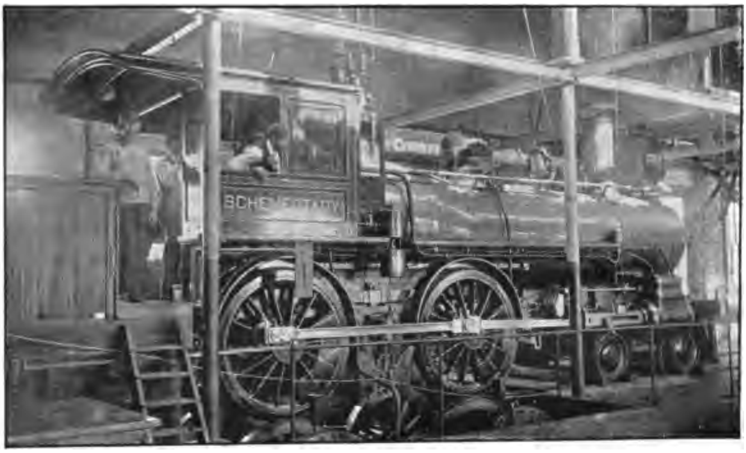

FIG. 5.—The Locomotive in the Laboratory, 1891–94.

Fig. 5 from a photograph is a view of the locomotive in place upon the plant; Fig. 6 shows a complete general view, in elevation, of the locomotive and its mounting machinery; and Fig. 7 shows in plan the mounting machinery only.

Reference to these figures will show that there was a heavy

rubble foundation, capped at convenient points with cut stones rising 10 inches above grade-line. Upon these stones were placed well-seasoned oak timbers arranged in two lines, each composed of three lengths, 4×14 inches. The timbers of each line were well bolted to each other and were securely anchored to the foundation. Upon the timbers rested the bearings of the supporting axles, which were thus given 14 inches of oak to constitute an element of elasticity between them and the foundation.

The supporting wheels were of the same diameter with the locomotive drivers, and similar in other respects save that the cranks and counterweights were omitted. Their faces were turned flat, with the inside edge rounded as in a rail.

The four friction-brakes which provided the load for the supporting shafts, and which are shown in position in Figs. 6 and 7, were designed on the principle developed by Professor George I. Alden and already described by him.* The details of the brake design under consideration will be given farther on, but it is important to state here that the principle as developed by Professor Alden provided extensive rubbing surfaces of cast iron and copper. Excessive wear was prevented by thorough lubrication. The intensity of the brake action was controlled by water pressure, by which means the rubbing surfaces were brought into contact more or less intimate, and the heat evolved was carried off by water circulation.

By reference to the plan and elevation, Figs. 6 and 7, it will be seen that there was no provision for measuring the load at the brakes, where, instead of a weighted lever, anchor-rods were used to secure the case of the brakes to the foundation. The value of the load appeared at the dynamometer connected with the draw-bar of the locomotive. The water supply for the brakes was furnished by a 3-inch pipe. It passed first a balanced valve *A*, around which there was a by-pass controlled by valve *B*. From the tee *C* the pipe was branched for the several brakes, 2½-inch piping serving for two brakes, and 2-inch for each individual brake. Valves, *D*, were provided in the supply-pipe for each brake, so that any one might be entirely cut out or have its action modified to any desired extent. The water from each brake was returned by a separate pipe to a point *E*, where valves were provided by which the amount of water allowed to pass each brake was regulated. From these valves the water

* Transactions of the American Society of Mechanical Engineers, Vol. XI, p. 959 *et seq.*

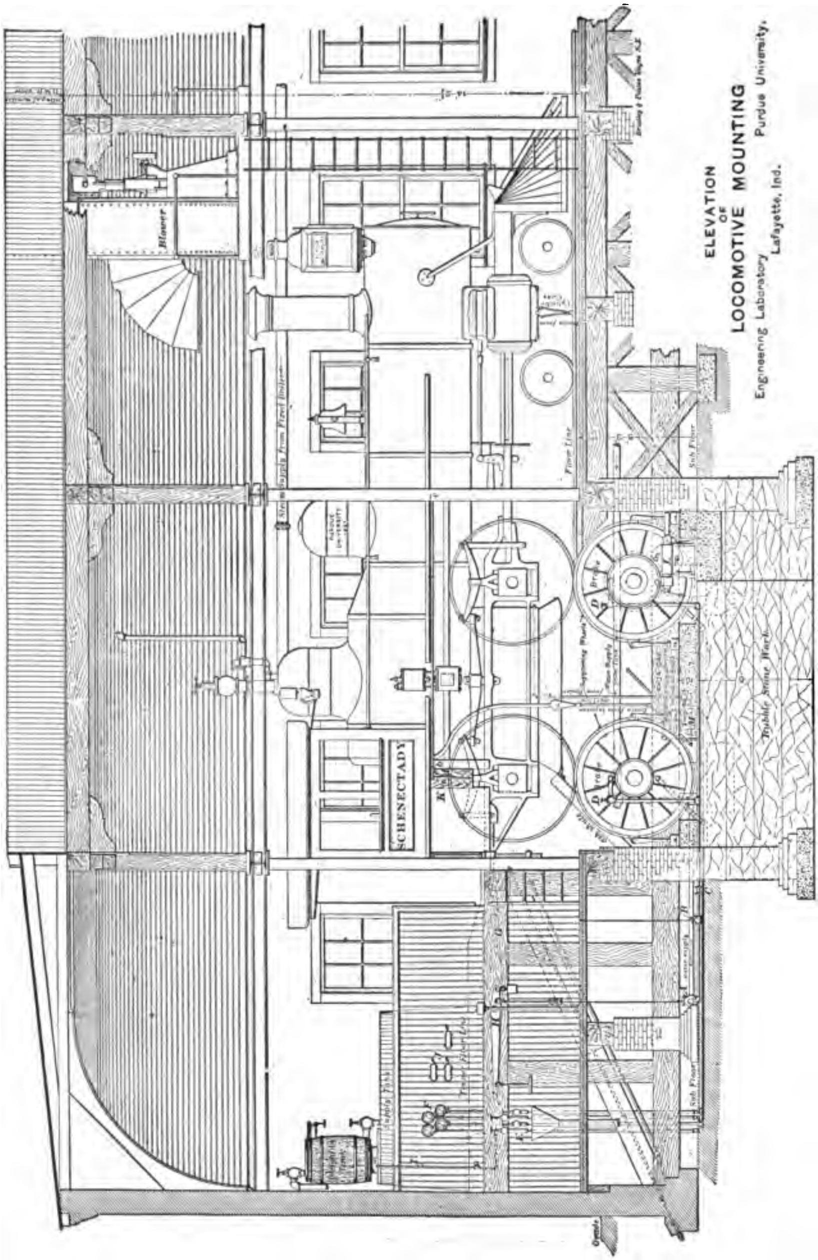
ELEVATION
OF
LOCOMOTIVE MOUNTING
Engineering Laboratory Purdue University,
Lafayette, Ind.
Blower
SCHENECTADY
Floor Line
Sub Floor
Supporting Wheels
Brake
Rubble Stone Work

FIG. 6.

flowed in an open stream and was finally discharged into a sewer. The water pressure within each brake was indicated by one of the four gauges at *F*, Fig. 6.

The balanced valve *A* had its spindle connected with one of the levers of the dynamometer in such a way that its position was controlled by the pull, or if backing, by the push exerted by the locomotive. Thus, suppose the locomotive to be in motion and the outlet-valves at *E* adjusted to allow the passage of enough water to keep down the temperature of the brakes; suppose also that the pull of the locomotive were such as to bring the weighted lever of the dynamometer to its mid-position, then there would be a definite opening of the balanced valve, and a definite water pressure within the brakes would result. If now, for any reason, the weighted lever should fall, there would be a corresponding increase in the opening of the balanced valve and, hence, an increase of water pressure within the brakes. The greater pressure would result in greater resistance to the movement of the supporting wheels and, hence, in a stronger pull of the locomotive on the dynamometer, and this increased pull would tend to lift the weighted lever again. Similarly, if for any reason the pull of the locomotive were sufficient to raise the lever beyond its central position, the balanced valve would respond by reducing the water pressure within the brakes; the tractive force of the engine would then decrease and the dynamometer lever would fall. When, therefore it was desired to increase the load on the locomotive it was necessary only to place the additional weight upon the lever of the dynamometer, and the corresponding increase in the load was furnished automatically by the brakes. By a proper adjustment of the by-pass valve, *B*, the lever could be made to stand exactly in its central position.

The traction dynamometer was made up of a system of levers. The first lever in the system, shown by dotted outline at *G* in Figs. 6 and 7, had direct connection with the locomotive draw-bar. The last lever, shown at *H*, carried an ordinary weight-holder. The whole arrangement was such that, whether the engine moved ahead or aback, the stress was transmitted by the draw-bar, and its value shown by the weight necessary to balance the lever *H*.

The dynamometer levers were carried by a heavy framework, which was well secured to the locomotive foundation and to surrounding parts of the building. The character of the framing is but imperfectly shown by the drawings.

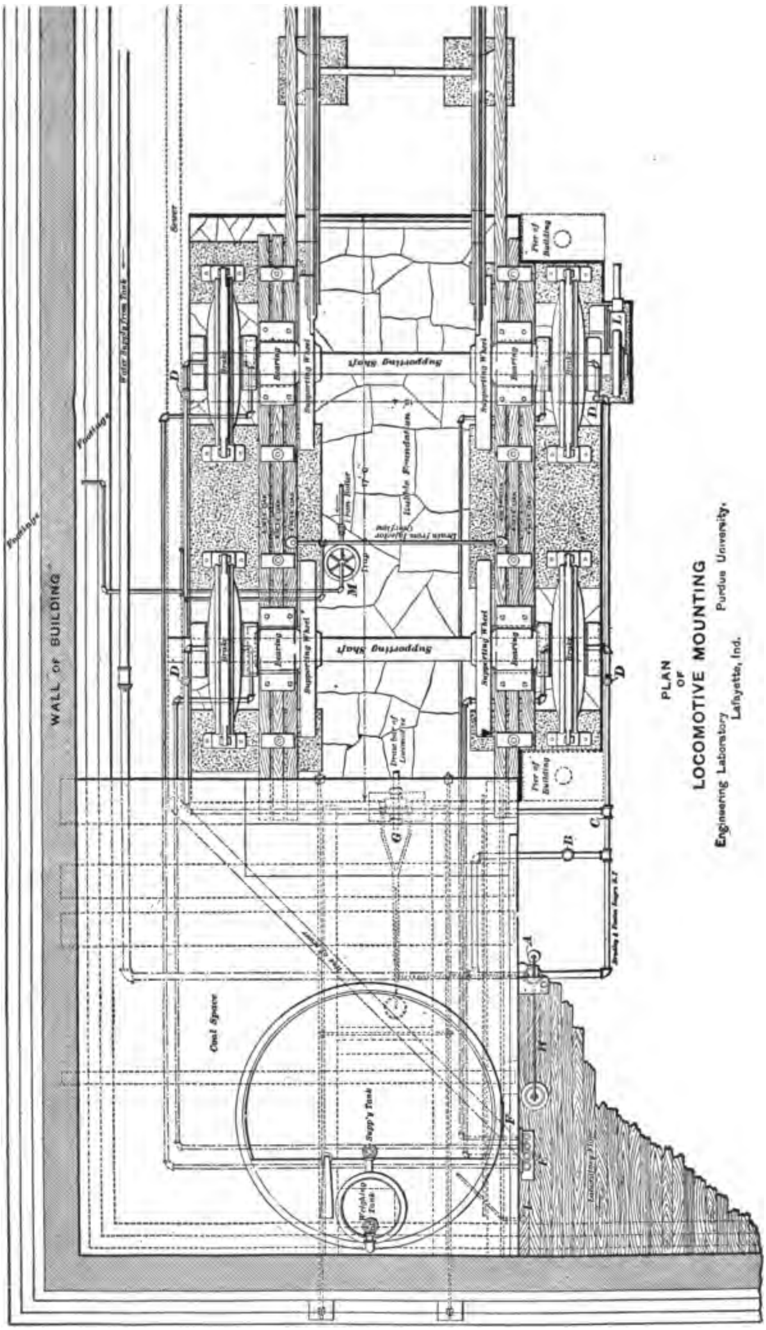
PLAN
OF
LOCOMOTIVE MOUNTING
Engineering Laboratory
Purdue University.
Lafayette, Ind.
WALL OF BUILDING
Footings
Water Supply from Tank
Sewer
Supporting Shaft
Supporting Wheel
Bearing
Brake
Isolated Foundation
Drain from Injector
Coal Space
Supp'y Tank
Weighing Tank
Pier of Building

FIG. 7.

While the draw-bar was the only active agent by which the horizontal movement of the locomotive was controlled, there was ample provision of chains and buffers to check any excessive movement which might occur.

Above the levers of the dynamometer a floor was laid which chiefly served the purposes of a tender. It gave room for the storage of a limited quantity of coal, and for a tank from which the locomotive injectors drew their supply. Connected with the water-tank was a glass gauge, *I*, and above the tank a weighing-barrel through which the tank received its supply. Scales for weighing fuel were also given a place on the "tender floor."

The three counters at *J* were connected, respectively, with the rear driving-axle and with each of the two supporting shafts. These gave a ready means for determining the speed of the engine, and the per cent of slip between the drivers and their supporting wheels.

The telltale at *K* showed the position of the locomotive relative to the supporting wheels. The board *a* was fastened to the locomotive and consequently moved with it; the rod *b* was connected at one end, *c*, to an iron column as a fixed point, and at the other end to the pointer *d*. This pointer was pivoted to the board *a* at *e*, so that any backward or forward movement of the locomotive was greatly multiplied in the similar movement of the lower end of the pointer *d*.

A tangent-wheel and screw were provided at *L* for the purpose of turning by hand the forward supporting axle and hence the engine, whenever it might be desired to do so, as, for example, for convenience in valve-setting. When not in use, the screw could be disengaged.

The truck-wheels of the engine rested upon light rails which were fixed at the level of the laboratory floor and extended in front of the engine a distance sufficient to allow the whole machine to be moved forward off the supporting wheels, whenever the latter needed to be take out for repairs.

A Sturtevant $4\frac{1}{2}\times6\frac{3}{4}$ steam-blower, located above the engine (Figs. 6 and 8) but not in pipe connection with it, removed from the room everything given out by the locomotive stack, without changing, materially, the draft conditions under which the locomotive worked.

The cylinder-cocks and the overflow-pipes from the injectors

were all in loose connection with the sewer. The discharge from the overflow-pipes could be directed into weighing-barrels.

The boiler was in pipe connection with the fixed boiler which supplied steam for general use in the laboratory, so that the locomotive might be used to supply steam to other apparatus, or the fixed boiler to supply the locomotive. In the latter case the locomotive boiler was drained by the steam-trap M; by this means the boiler was freed from water, and as a consequence all its parts were kept at the same temperature. The greater convenience attending


Stack.

FIG. 8.—The Blower above the Engine.

the operation of the fixed boiler made its use desirable when problems were studied which affected only the mechanism of the engine.

The following is a description of some of the more important parts making up the plant, the details of which are not clearly shown by the drawings thus far referred to.

5. The Supporting Axles were of hammered iron and of the form and dimensions shown by Fig. 9. The bearings for these shafts were 8″ in diameter and 16″ long. Each bearing was fitted to a cast-iron plate 14″×36″×2″, which plate, in turn, was carefully bedded upon the oak timbers (Figs. 6 and 7). The whole was made secure by four bolts which passed through the bearing, plate, and timber.

6. The Alden Friction-brakes, which supplied the load to the supporting axles and, hence, to the locomotive itself, are shown in position by Figs. 6 and 7, and in detail by Fig. 10. They were designed and constructed with the consent of Professor Alden, to whom the author is indebted for many courtesies.

The cast-iron moving disk *K*, Fig. 10, Sec. AB, is keyed to the supporting axle (not shown), and hence turns with it. The power of the locomotive is transmitted by the supporting wheels and axle to the disk, on either side of which are light copper plates, *L*, which form a part of the enclosing case. The sides *M* of this case have

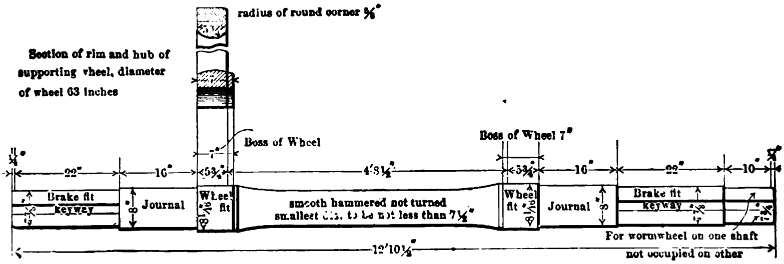

FIG. 9.—Supporting Axles.

bearing on the hub of the moving disk, and are connected around their outer edges by the distance-ring *N*, and by through-bolts (Sec. CD). The copper plates already referred to are clamped at their outer edges between the sides of the case and the distance-ring *N*. At their inner edges the joint is made by means of wrought-iron rings, secured by closely spaced machine-screws tapped into the case. The copper plates are thus held very near to the moving disk, but they are not necessarily in absolute contact with it. Between the copper plates and the sides of the case are annular spaces within which water may be circulated. It will be seen that the case with its copper plates is quite free to revolve upon the hub of the cast-iron disk; or if, as in practice, the case is at rest, the disk may revolve freely within it. It will be seen, also, that water under pressure in the annular spaces will force the light copper plates against the moving cast-iron disk; that, as a result of this contact, there will be a tendency on the part of the case to turn with the cast-iron disk, and that if this tendency is overcome by an outside resistance, power will be absorbed. This, in brief, is the action of the brakes; the case, in the present instance, is prevented from turning by anchor-rods connecting it with the foundation (Fig. 10).

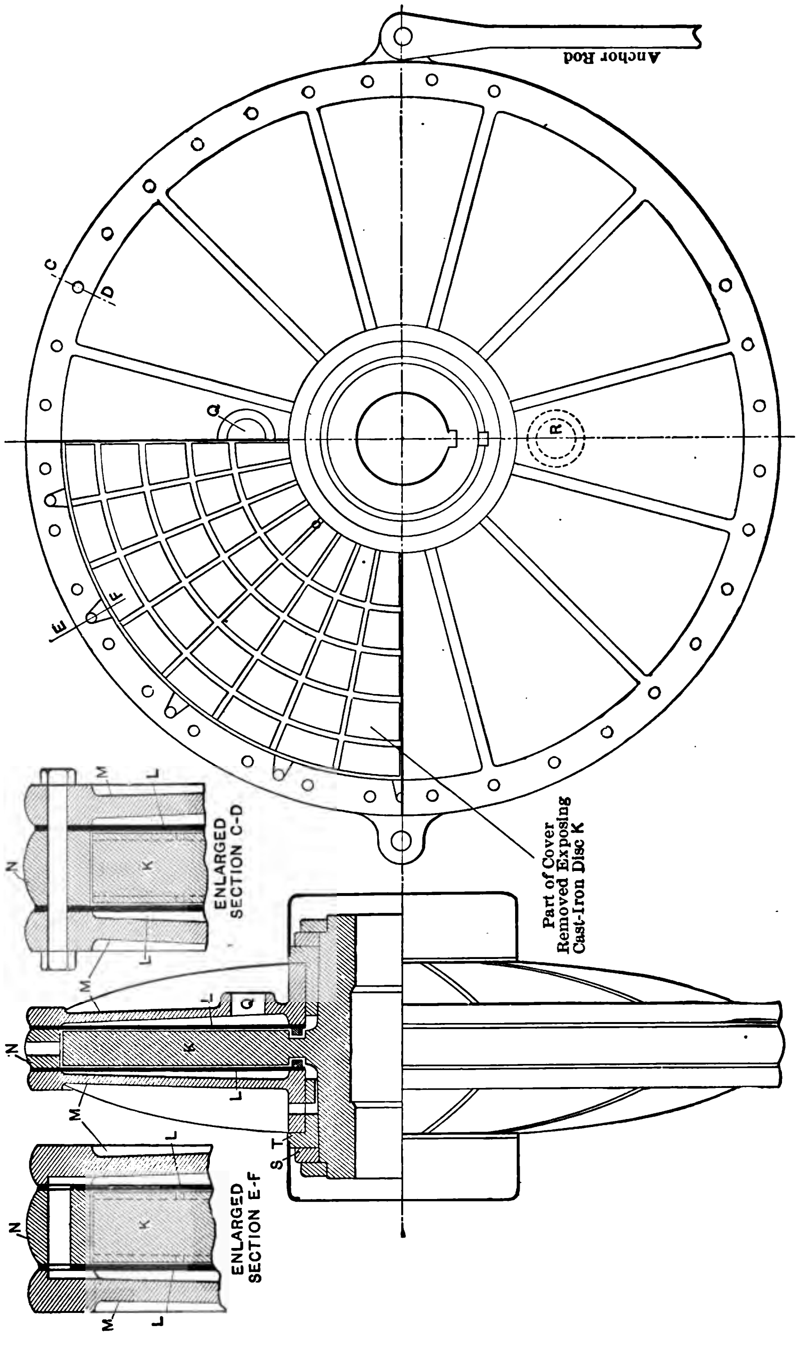

FIG. 10.—Alden Friction-brake.

The circulating water enters at the opening Q (Fig. 10), passes from the annular space on this side of the case to that on the other side by eighteen $\frac{3}{8}$-inch holes through the copper plates and the distance-ring (Sec. EF), and is finally discharged at the opening R, sufficient water being allowed to pass to carry away the heat resulting from the friction. To prevent the water pressure within the brake from spreading the sides of the case, the rings S are fitted over a feather to the hub of the moving disk, and are held to their place by nuts which screw up to a shoulder.

When the brake is in use, all clearance space between the copper plates and about the moving cast-iron disk is filled with oil. The distribution of the oil is secured by thirty-two radial grooves on each face of the cast-iron disk (Fig. 10), and by a spiral groove

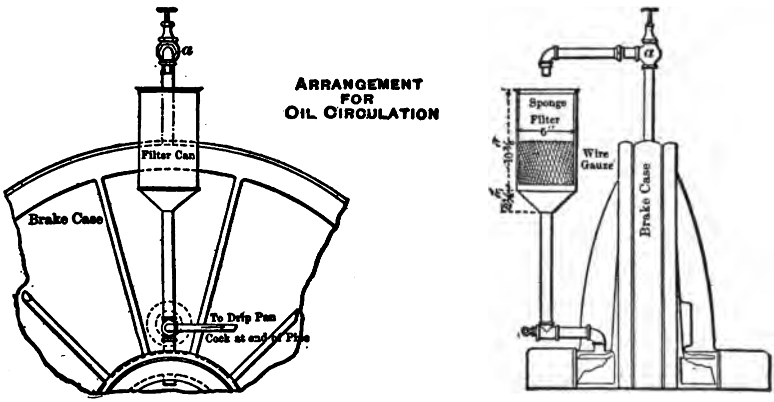

FIG. 11.

extending from the inner edge to the circumference of the rubbing surfaces of the disk, with a pitch of about 4 inches. The entire rubbing area of the cast-iron disk is thus split up into surfaces the length or breadth of which in no case greatly exceeds 4 inches. The spiral oilway gives these surfaces such positions that in passing a given point on the copper plate they continually change their alignment, which fact, it is believed, greatly assists in the distribution of the oil. The radial grooves give rise to a slight pumping action, by means of which the oil may be kept in circulation between the center and circumference of the brake. Provision for this circulation is made as shown by Fig. 11; the oil passes the valve and piping a from the highest point in the brake, is received by the filter-can,

and is thence delivered to the center of the brake. The filter-can also serves the purpose of a supply reservoir, and always contains surplus oil when the brake is in action. This circulation helps to maintain the oil at a uniform temperature, subjects all parts to the same service, and gives a ready means for detecting any defect which may arise in the lubrication of the brake. One of the brakes with one-half of the case removed, exposing one of the rubbing surfaces of the moving disk, is shown by Fig. 2.

As preliminary to the design of these brakes, the load which they would need to carry was determined as follows: The locomotive drivers were 63 inches in diameter, and the sum of the moments about the two driving-axles when the engine was exerting a tractive force of 16,000 pounds (assumed to be maximum) was therefore $16,000\frac{63}{2\times 12}=$ 42,000 foot-pounds, which, since the supporting wheels were of the same diameter with the drivers, was the moment under which the brakes on the supporting axles must work. Other considerations led to an early decision to use two brakes on each axle, making it necessary, therefore, for each brake to act under a maximum moment of 10,500 foot-pounds.

FIG. 12.—A Friction-brake with One-half of Case Removed.

Before fixing upon the dimensions of the proposed brakes, a rather extensive series of experiments was made upon a small Alden brake then in use in the laboratory, for the purpose of determining the probable value of the coefficient of friction which would attend the action of such a brake. These preliminary experiments were made on a 21-inch disk-brake while driven at speeds varying from 300 to 450 revolutions per minute. From these experiments it appeared that lubrication could be maintained with certainty under a water pressure of 40 pounds per square inch, and this pressure was adopted as the maximum to be used in the design of the larger brakes herein described. It also appeared that the *apparent coefficient of friction* * varied from 2.7% to over 4%, depending largely on the

* By apparent coefficient of friction is meant that factor which is obtained by assuming that the entire moment of the brake is due to the friction between the rubbing surfaces of the moving disk and the copper plates. Since there are other rubbing surfaces, it is clear that the apparent coefficient of friction is larger than the actual coefficient.

viscosity of the oil used and upon the temperature of the brake. It was thought that 3.5% would be a safe coefficient for moderate speeds, and this factor was accordingly used.

The moment in foot-pounds required to revolve one disk upon another against the action of friction, when the two are bearing face to face, is the product of the area in contact, the pressure of contact per unit area, the coefficient of friction, and two-thirds the radius of the disk in feet. Representing this statement by an equation, we write

$$M = Apf\frac{2}{3}\frac{r}{12} = \pi r^2 pf\frac{1}{18}r$$

$$= \frac{\pi}{18}pfr^3,$$

where M = moment of force in foot-pounds;
A = area in contact in square inches
$= \pi r^2$;
p = pressure in pounds per unit area;
f = the coefficient of friction;
r = the radius of the circular plate in inches.

But each brake of the design in question (Fig. 10) consisted of a revolving disk with a fixed copper plate on either side, thus giving two rubbing surfaces, and for such a brake the moment will be

$$M = \frac{\pi}{9}pfr^3. \quad \ldots \ldots \ldots \quad (1)$$

Again, the copper plates forming the rubbing surfaces were not complete disks, but rings. The moment necessary to revolve such a disk against such a ring will be the same as the moment necessary to revolve a disk of a radius equal to the outer radius of the ring, less the moment necessary to revolve a disk of a radius equal to the inner radius of the ring. Now if

r_1 = the outer radius of the ring in inches,
r_2 = the inner radius of the ring in inches,

we may write, by the aid of equation (1), for the moment of the ring in foot-pounds,

$$M = \frac{\pi}{9}pfr_1^3 - \frac{7}{9}pfr_2^3$$

$$= \frac{\pi}{9}pf(r_1^3 - r_2^3). \quad \ldots \ldots \ldots \quad (2)$$

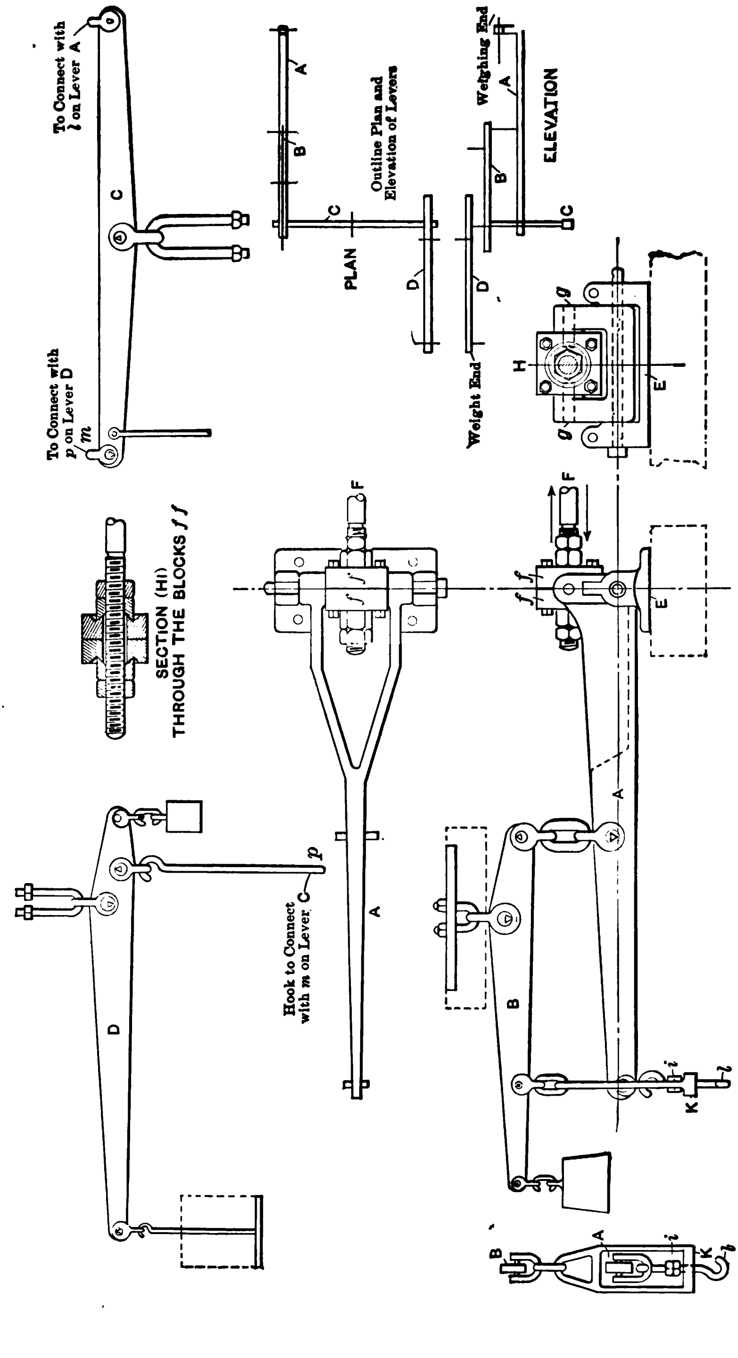

Fig. 13.—Locomotive Dynamometer.

As already stated, previous experiments with a trial brake had shown that a coefficient of friction of 0.035 could be depended upon. The maximum water pressure assumed for the design was 40 pounds per square inch, and trial solutions which were carried out as the design was in process of development made r_1 equal to 28 inches and r_2 equal to 10 inches. Substituting these values in equation (2) gives

$$M=\frac{3.14}{9}\times 40\times 0.035\times(21952-1000)=10234,$$

which is to be compared with the required moment of 10,500 foot-pounds. It is evident that the best available information indicated that the dimensions chosen were such as to make the brakes well suited to the requirements of their anticipated service.

7. **Traction Dynamometer** is shown by Fig. 13. The outline plan and elevation in this figure show the relative position of the different levers when in place. The details of each lever are shown by the remainder of the plate.

The main lever A being first referred to, the following description will be seen to apply. This lever was supported by a round steel pin which connected it with the cast plate E, and which in turn was securely bolted to the heavy timbers which formed a part of the framework behind the locomotive. The round rod F was an extension of the locomotive draw-bar, and the pull, or push, of the locomotive was exerted along the line of its axis. Stress was transmitted from this rod to the blocks f by nuts as shown, the fit between the nuts and the blocks being spherical (detailed view, Fig. 13) to allow slight changes in the direction of F. The blocks f were connected with the short arm of the lever A by the round steel pins g, and the long arm of this lever was engaged by the hook l, which was free to move within the link k. To illustrate the action of this part of the dynamometer, let it now be assumed that the pull of the locomotive on the draw-bar F is ahead, that is, in the direction of the upper arrow; then the stress on the draw-bar will result in a tendency to raise the hook l, and the lever B with its weight will serve only as a counterbalance to the lever A. But if it be assumed that the locomotive is working aback, that is, that the stress in the draw-bar is in the direction of the lower arrow, the long arm of the lever A with the hook l will tend to fall. Motion of l in this direction, however, is soon arrested by the link k, which by virtue of its

connection with the lever *B* rises as *l* falls, until it engages the check-nuts *i*; these are thus made a means of transmitting the stress to the hook *l*, which as before will move upward, while the shackle at the upper end of the hook *l* is entirely free from stress. It will thus be seen that, whether the locomotive is assumed to be working in forward or in backward gear, its tractive force is made manifest by an upward movement of the hook *l*, from which point two simple levers complete the dynamometer.

The direction of the lever *C*, which connected with hook *l*, was at right angles with that of the lever *A*, its purpose being to bring the last lever *D* of the system out from behind the locomotive. The lever *D* carried a weight-holder arranged to receive twenty 10-lb. weights. The ratio of the whole system was as 1 to 100; each weight, therefore, that was balanced at the weight-holder represented a tractive force exerted by the locomotive of 1,000 lbs.

A small dash-pot (not shown) was attached to the last lever, *D*. The lever *C* had depending from it a light rod, which controlled a balanced valve *B* (Figs. 6 and 7) in the pipes upplying the brakes with water. By means of this valve, as previously described, the load on the brakes was made to vary automatically with the position of the last lever of the dynamometer.

The locomotive while in motion could be given a final adjustment to its place on the supporting wheels by means of the nuts on the draw-bar.

Having now examined the more important principles underlying the design of the first plant, we may make inquiry concerning its behavior in service.

8. Behavior of the Mounting Mechanism of the First Plant.— Any considerable arrangement of machinery which is new in the details of its design, or which is required to serve purposes which are new, must for a time be considered experimental. The mounting mechanism of the locomotive testing-plant was novel in many of its details, and the conditions surrounding its use were new. Each element entering into the make-up of the plant, therefore, demanded in its turn its full share of patiently-bestowed and long-continued attention. The most important element was the friction-brakes by which the power of the locomotive upon the mount was absorbed. These, as already stated, had been designed upon a principle which had been previously developed by Professor Alden, but no brakes had before been made approaching in capacity those of the Purdue plant.

Their ability to perform the work for which they were intended was early settled beyond question. When they were first used, the oil became very hot about the outside of the case, while that in other parts of the apparatus remained cool. To avoid this accumulation of heat around the outside of the case, closed circulating pipes were added to convey the hot oil from the outer portions of the case back to the center, the radial curves in the disks being sufficient to maintain a pumping action by which there was a continuous flow of oil from the outside to the center of the brake. These pipes took the place of the open circulation provided for in the original design. They served, also, to maintain all portions of the brake at an approximately uniform temperature, the excess of heat being carried away by the circulating water as already described. The brakes were lubricated by a cheap grade of cylinder-oil, which was supplied by the cans shown by Fig. 11. There was always some leakage at the center, but no more than was desirable to insure lubrication of that point. The drip was caught and returned to the can.

In the original design of the brakes it was assumed that the work absorbed would be proportional to the water pressure exerted upon the copper plates, an assumption which would be true provided the lubricant between the copper plates and the cast-iron moving disk was always in the same condition. In practice it appears that the work is all done on the oil and any change of condition resulting in a change in the amount of work absorbed produces a corresponding effect upon the temperature and, hence, upon the viscosity of the oil. Other things being equal, therefore, increasing or diminishing the pressure does not result in a proportional increase or diminution of the amount of work absorbed. The effect of this observation is expressed by saying that the amount of work absorbed by the brakes depends quite as much upon the temperature of the oil in the brakes as upon the pressure exerted between the copper plate and the moving disk; and in practice, excepting when the speed of revolution is very small, it is found best to use a moderate water pressure, and to vary the load by varying the brake temperature, this being easily accomplished by controlling the amount of circulating water. The water pressure under ordinary conditions of service rarely exceeds 10 pounds.

It has been stated that the work absorbed by the brakes is practically limited to that which is represented by the force necessary to overcome the viscosity of the oil. Evidence of the truth of this

statement is to be found in the fact that the bearing pressure between the copper plate and the moving disk, being seldom over 10 pounds per square inch, causes no perceptible wear of the rubbing surfaces so long as lubrication is maintained. The copper plates and the cast-iron disk which constitute the rubbing surfaces of the four brakes, after a total of six million revolutions show wear in only in spots on the copper plates where, owing to imperfections in the surface, small areas have received a concentration of pressure. Since the work absorbed by the brakes depends largely upon the temperature of the oil which lubricates their rubbing surfaces, it will appear that the

FIG. 14.—The Dynamometer, Mechanism for Controlling Pressure on Brakes, and the Revolution Counters. First Plant 1891.

maximum load will be greatest when the heat developed between the rubbing surfaces is most quickly conducted away. The copper plates in the brakes under consideration were made $\frac{3}{16}$ of an inch in thickness. It is probable that less cooling water would be required and the action of the brakes would be improved if the thickness of these plates were considerably reduced. They can perhaps be reduced until only a sufficient amount of metal remains to withstand the shearing forces to which the action of the brake subjects them.

The dynamometer to which the draw-bar of the engine was connected (Fig. 13), while reasonably satisfactory, was not well calculated to absorb the heavy vibrations which at high speed the locomotive brought upon it. The structure behind the locomotive was

several times strengthened. First, the woodwork upon which the dynamometer was mounted was reënforced, the anchorage behind it increased, and a small dash-pot added to its last lever. Later a larger dash-pot having a diameter of 12 inches was interposed between the fixed tender-frame behind the locomotive and a bracket attached to the locomotive foot-plate. This, being entirely independent of the dynamometer, served to absorb the larger part of the vibrating forces incident to high speed, and after its addition no difficulty was experienced either in holding the engine or in measuring approximately its pull. The dynamometer was not, however, a piece of apparatus whose accuracy could be relied upon under all conditions of service.

In view of the enormous force which a locomotive is capable of exerting it would appear, at first sight, that an error of 50 or even 100 pounds in the determination of draw-bar stresses would be of slight consequence, and that great accuracy in this matter would not be required. Under some conditions this conclusion is correct, but under others it is far from true. The work done at the draw-bar is the product of the force exerted multiplied by the space passed over. If the force exerted is great and the speed low, a small error in the draw-bar stress is not a matter of great importance, but if the reverse conditions exist—if the force is small and the speed is high—then it is absolutely necessary that the draw-bar stresses be determined with great accuracy. Moreover, high speeds necessarily involve low draw-bar stresses. A locomotive which at ten miles an hour may pull 12,000 pounds will have difficulty, when running at sixty miles an hour, in maintaining a pull of 2,500 pounds. When, therefore, in the general progress of events, the opportunity came to improve the character of the dynamometer, considerations such as have been stated were deemed of sufficient importance to justify the purchase of the most accurate apparatus which could be found.

The exhaust-fan over the engine (Fig. 8) was a detail made necessary by the character of the room in which the locomotive was located and the variety of equipment by which it was surrounded. It never failed to serve its purpose, and when steam was being raised it was invaluable.

A minor difficulty encountered in the course of the early work upon the plant was that due to the tendency of oil used upon the bearings of the supporting axles to follow the arms of the supporting wheels out to the rim and thence to reach the locomotive drivers.

This oil by spreading over the faces of the supporting wheels and drivers reduced adhesion and greatly interfered with the action of the plant. A remedy was found in the attachment of a ring to the face of the supporting-wheel hub of such form as would receive and retain by centrifugal force all drip occurring during a run. Still other minor changes will be found worked out as accomplished facts in Purdue's second plant, a description of which is to follow.

FIG. 15.—The Engineering Laboratory, Purdue University, prior to Jan. 23, 1894.

9. The Work of the First Plant.—During the school year of 1891–2, following the installation of the plant, work upon it was directed more to the perfection of mechanical features than to the acquistion of scientific data. Nevertheless, during this year twenty efficiency tests were run, many of them at light power and almost all with the throttle only partially open, and later an experimental investigation concerning the action of the counterbalance was undertaken.

It was not until the fall of 1893 that the plant was in such complete working order that tests at high power could be run with certainty. During the latter part of this year, however, a considerable number of such tests were carried out and much was expected from an early study of the results, but no essential fact was ever established from these data. On January 23, 1894, the Engineering Laboratory, which by a succession of additions had increased greatly in size, was burned. All experimental data which had not been published were lost, and the locomotive went down in the wreck. The fire entailed a heavy burden of labor and expense. But with

the new responsibilities which it brought there came also new opportunities. All details of the mounting mechanism were most carefully reviewed, and every fragment of experience was made to serve a useful purpose in the design of a new plant. A permanent track 8,000 feet in length was laid to connect the laboratory with the railways of the country. The damaged locomotive was extricated from the ruin, sent out over the track and thence to Indianapolis for repairs. Upon its return a few weeks later, it was put under its own steam and backed in over the Purdue track directly to its place upon the supporting wheels of a new testing-plant. The ease and rapidity

FIG. 16.—The Locomotive-testing Plant Immediately after the Fire.

with which this trip was made were in striking contrast with the laborious methods which attended its first trip across the same territory. Four months after the fire the new work had been completed and the reconstructed engine was in position.

10. **The Second Testing-plant.**—The new plant, as shown by the accompanying plates, occupies a building especially planned to receive it, and is arranged for the accommodation of any locomotive. The dotted outline in Fig. 17 is to the scale of the University's locomotive Schenectady.

11. **The New Wheel Foundation.**—By reference to Figs. 17 and 18 it will be seen that there is provided a wheel foundation of nearly twenty-five feet in length. This is more than sufficient to include the driving-wheel base of any standard eight-, ten-, or twelve-wheeled

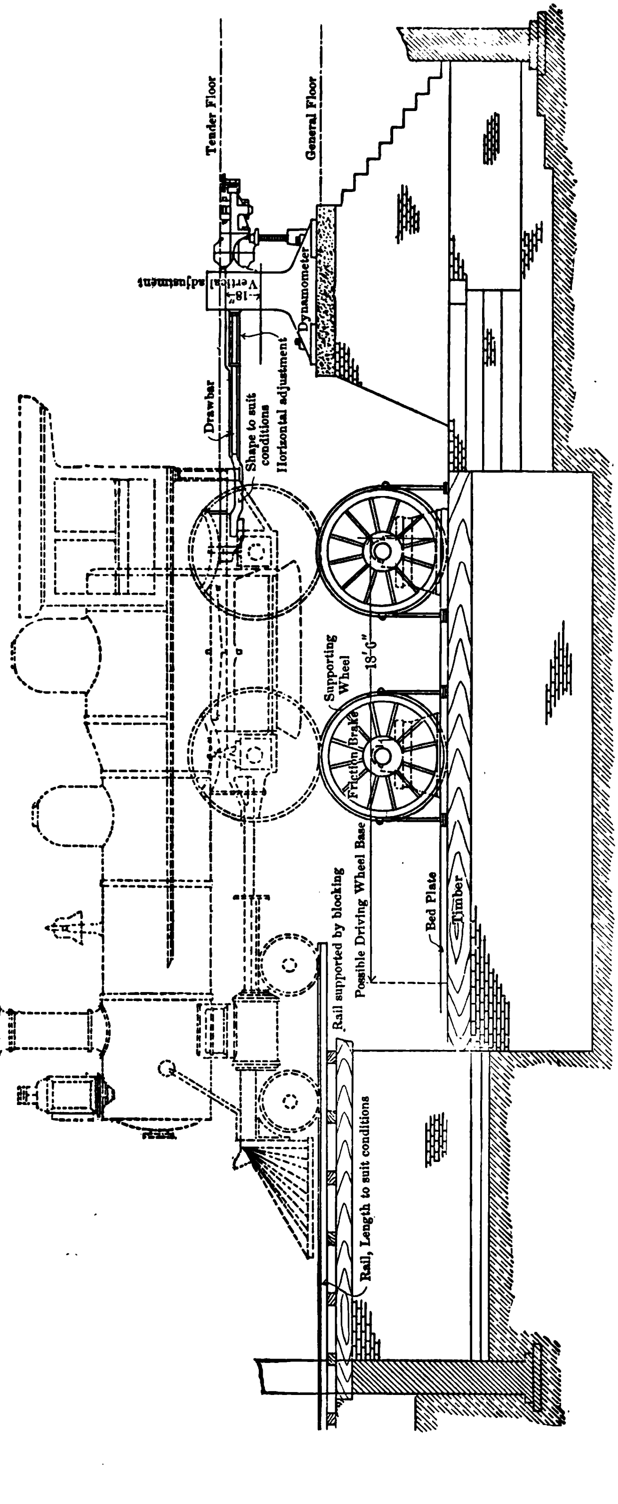

FIG. 17.—An Elevation of the Second Testing-plant, Purdue University, 1894.

engine. For engines having six wheels coupled, a third supporting axle will be added to those shown, and for engines having eight wheels coupled four new axles, having wheels of smaller diameter than those shown, will be used.

The wheel foundation carries cast-iron bed-plates, to which are secured pedestals for the support of the axle-boxes. The lower flanges of the pedestals are slotted and the bed-plates have threaded holes spaced along their length. By these means the pedestals may be adjusted to any position along the length of the foundation.

The boxes in use at present are plain, babbitted shaft-bearings, and between each bearing and its pedestal a wooden cushion is inserted. A bearing has been designed, for use in some special experiments which provides for the suspension of the axle from the springs, but has not yet been used.

The outer edges of the wheel foundations are topped by timbers to which the brake-cases are anchored. The brakes which absorb the power of the engine are those which were used in the original plant, and which have been described. The fire caused no serious damage to this portion of the apparatus.

12. **The Emery Dynamometer.**—The vibrating character of the stresses to be measured makes the design of the traction dynamometer a matter of some difficulty. The dynamometer of the original plant consisted of an inexpensive system of levers attached to a heavy framework of wood, the vibrations being controlled by dash-pots. In the present construction wood as a support is entirely abandoned and a massive brick pier, well stayed with iron rods, has been substituted. The dynamometer connects with the draw-bar at the rear of the locomotive. It consists of the weighing-head of an Emery testing-machine, the hydraulic support of which is capable not only of transmitting the stress it receives, but also of withstanding the rapid vibrations which the draw-bar transmits to it. The apparatus is of 30,000 pounds capacity, and at the same time is so sensitive that one standing in front of the locomotive may press with his fingers upon its pilot and cause a deflection of the needle of the dynamometer.

As is well known, the arrangement of the hydraulic support of the Emery testing-machine permits the weighing-scale to be at any convenient distance from the point where the stresses are received. Figs. 17 and 18 show only the receiving end of the apparatus. The draw-bar connects with this apparatus by a ball-joint, which leaves

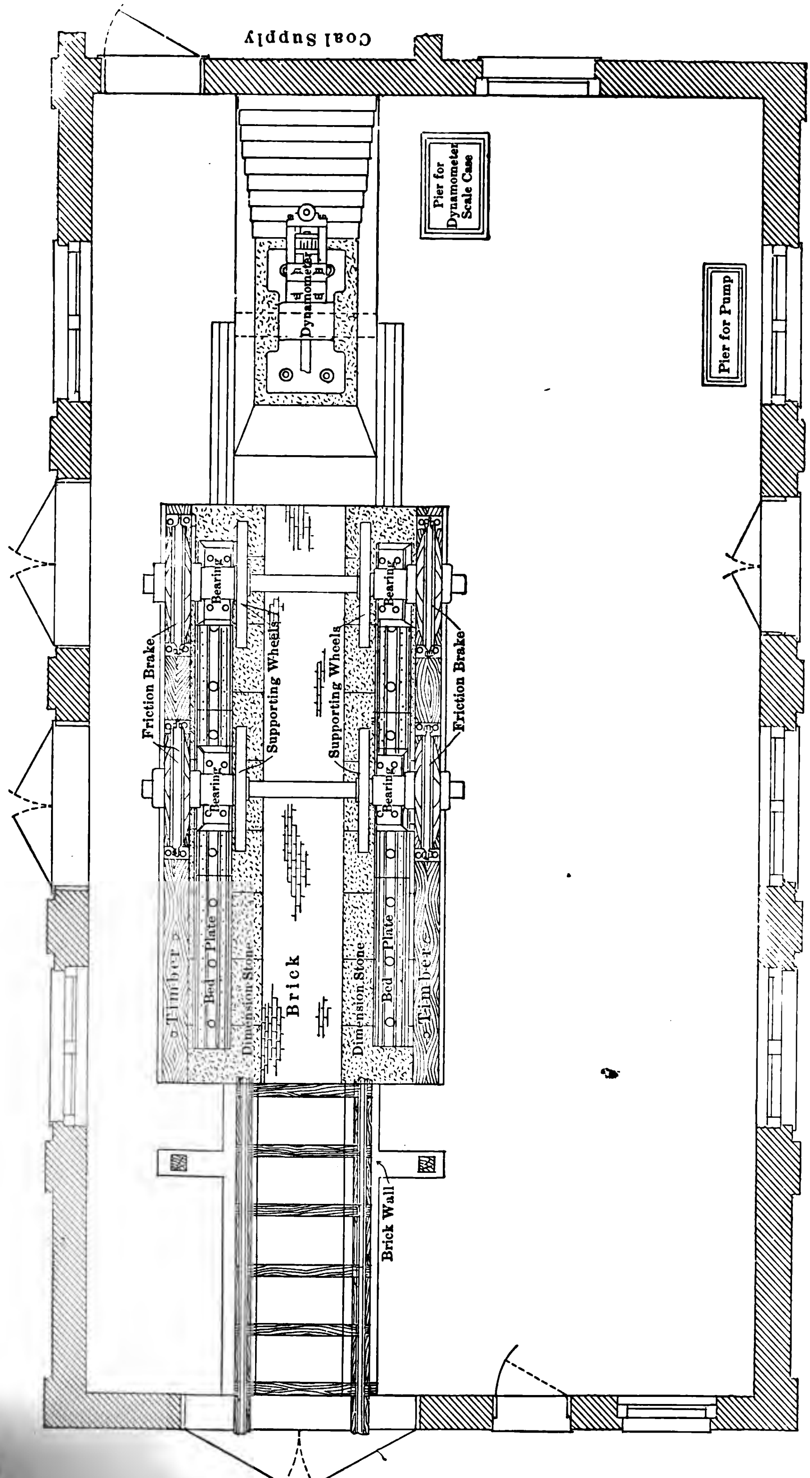

FIG. 18.—A Plan of the Second Plant, Purdue University, 1894.

FIG. 19.—The Dynamometer and Mechanism for Controlling Pressure on Brakes. Second Plant, 1894.

its outer end free to respond to the movement of the locomotive on its springs. A threaded sleeve allows the draw-bar to be lengthened or shortened for a final adjustment of the locomotive to its position upon the supporting wheels; and, finally, to meet the proportions of different locomotives, provision is made for a vertical adjustment of the entire head of the machine upon its frame.

13. **The Superstructure.**—Figs. 20 and 21 show the arrangement of floors. The "visitors' floor" (Fig. 21) and the fixed floors adjoining are at the level of the rail. The open space over the wheel foundation is of such dimensions as will easily accommodate an engine having a long driving-wheel base, movable or temporary floors being used to fill in about each different engine, as may be found convenient. The temporary flooring shown is that employed for the Purdue locomotive.

The level of the "tender floor" is at a sufficient height above the rail to serve as a platform from which to fire. At the rear is a runway leading to the coal-room, the floor of which is somewhat lower than the tender floor. A platform scale is set flush with the floor at the head of the runway. During tests the scale is used for weighing the coal which is delivered to the fireman.

The feed-water tank, from which the injectors draw their supply, is shown in the lower right-hand corner of Fig. 21. Above this supply-tank are two small calibrated tanks so arranged that one may be filled while the other is discharging.

The steam-pump shown on the visitors' floor is for the purpose of supplying water under pressure to the friction-brakes which load the engine.

The conditions under which the engine is operated are at all times within the control of a single person, whose place is just at the right of the steps leading to the tender floor. From this position he can see the throttle and reverse-lever and observe all that goes on in the cab. At his right is the dynamometer scale-case, wherein is shown the load at the draw-bar; in front are the gauges giving the water pressure on the brakes; and under his hand are the valves controlling the circulation of water through the brakes.

No attempt has been made in these drawings to show small accessory apparatus, neither does it seem necessary to give an enumeration.

14. **The Building.**—Fig. 23 presents several views of the locomotive building. The entrance-door, which opens upon the visi-

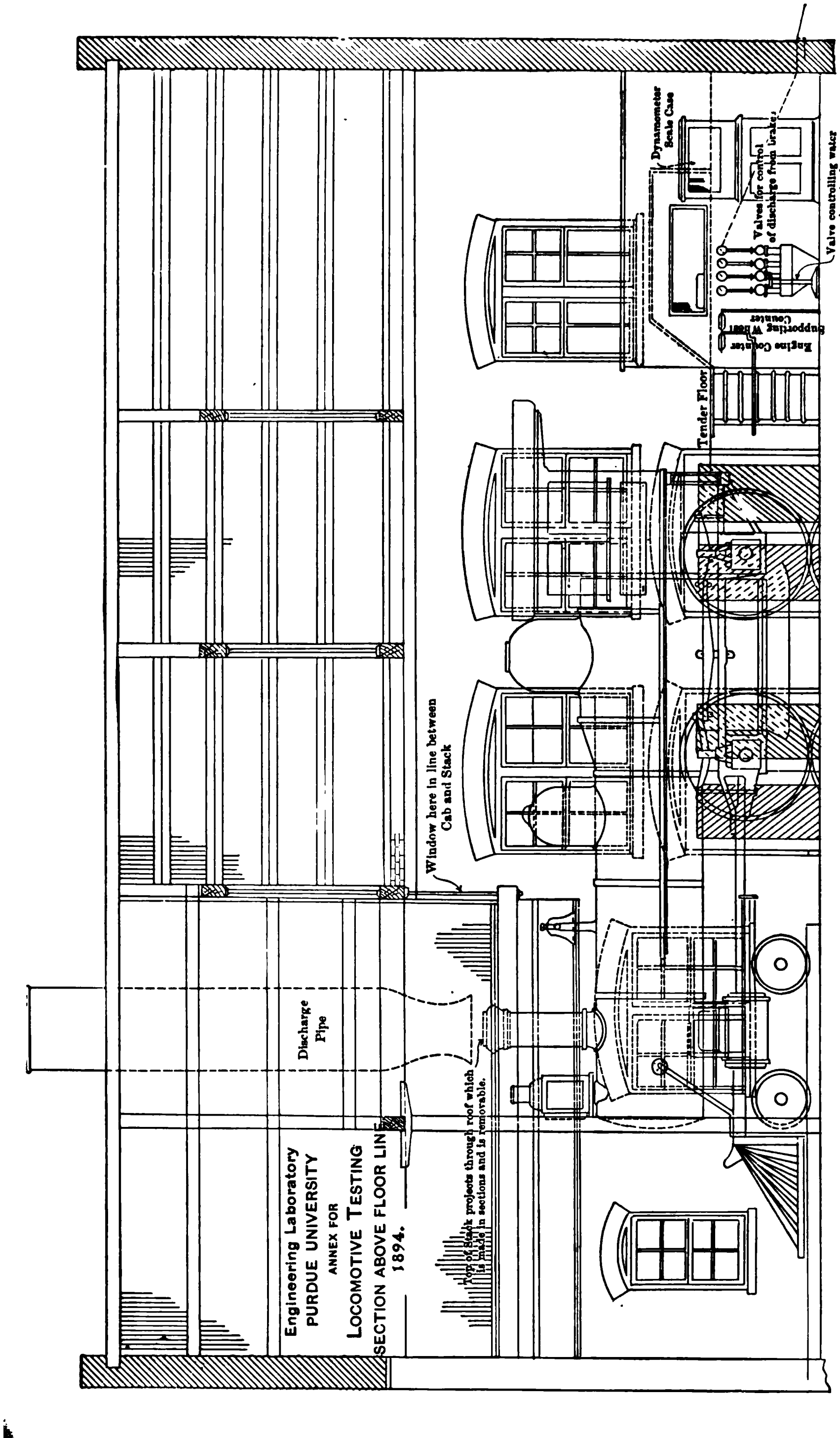

FIG 20.—Section of Building, Second Plant, Purdue University, 1894.

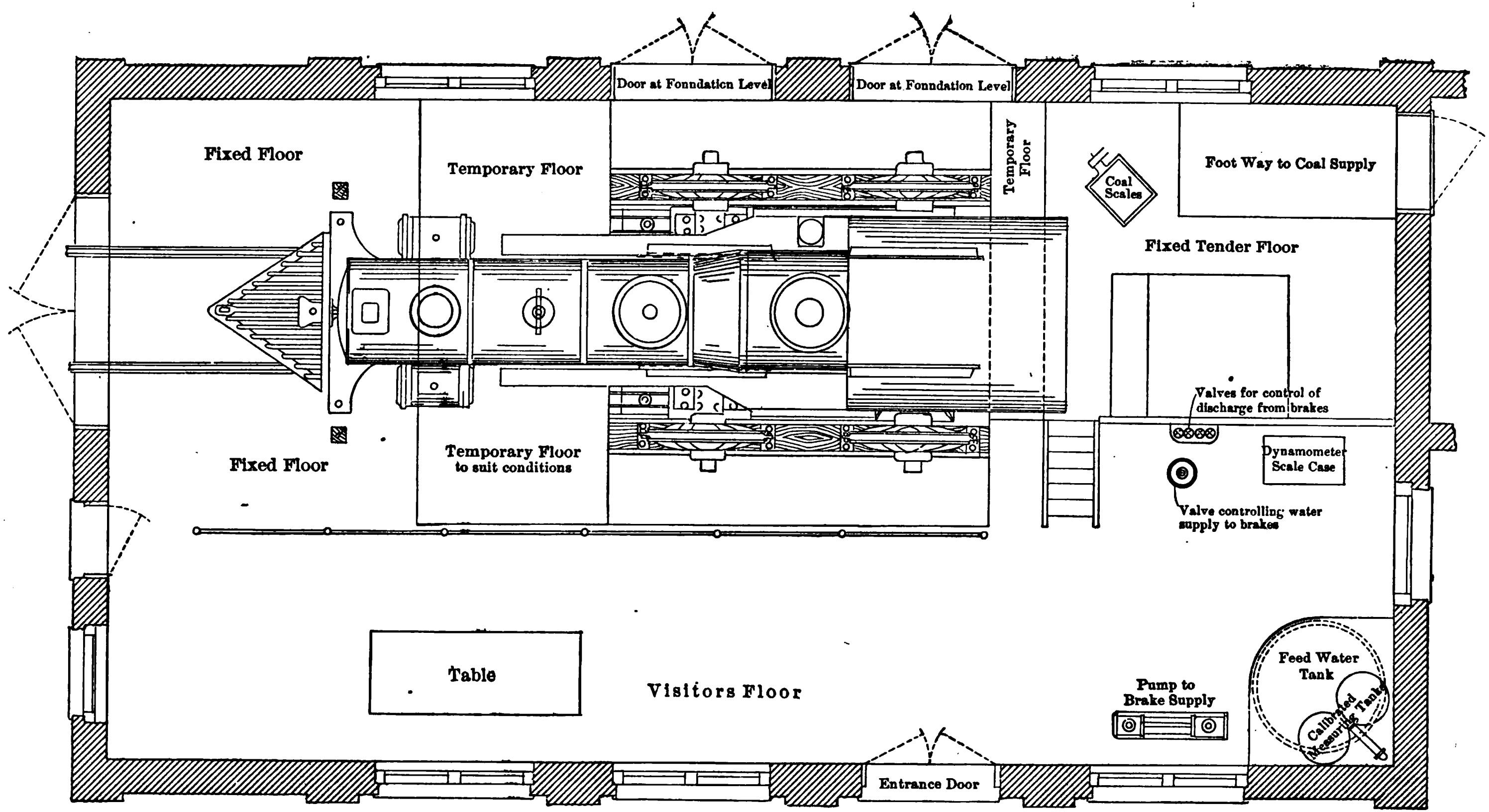

FIG. 21.—Floor-plan, Second Plant, Purdue University, 1894.

tors' floor, is shown in the south elevation. It is approached from the general laboratory, 150 feet away.

The north and west elevations show the roof construction by which the upper end of the locomotive stack is made to stand outside of the building. The roof sections shown may be entirely removed, and a door in the cross-wall, which extends between the removable roof and the main roof, provides ample height for the admission of the locomotive to the building. A window in this door (Fig. 20) serves to give the fireman a clear view of the top of the locomotive stack from his place in the cab, a condition which is essential to good work in firing. Above the stack is a pipe to convey the smoke clear of the building. To meet a change in the location of the stack this pipe may be removed to any position along the length of the removable roof. An illustration from a photograph of the interior, taken when the plant was in operation, is given as Fig. 22.

In connection with the plan of the building (Fig. 23) there is shown the arrangement of tracks for the locomotive and of those used for supplying coal. Fig. 24 is an exterior view of the locomotive laboratory when the plant is working, and Fig. 25 gives the position of the locomotive or annex laboratory, relative to the whole group of buildings of which it forms a part.

15. **Work with the New Plant** began in the fall of 1894 and continued without interruption for more than two years. In the course of this time most of the results were secured which are to be discussed in the succeeding chapters. Fifty or more efficiency tests of boiler and engine, under various conditions of speed, load, cut-off, steam pressure, and valve proportion, were run. A large amount of time was spent in determining the amount of work lost in machinery friction under different conditions of running. A study was made of the effect of high rates of combustion upon boiler efficiency, of the conditions affecting the draft action produced by the exhaust, of steam distribution by plain and Allan-ported valves, and of the power necessary to move balanced and unbalanced valves.

In the course of this work, and of that done upon the first plant, not less than 10,000 indicator-cards were taken, 20,000 miles were run, and 25,000 different observations made. From these data about 12,000 derived results had been obtained, making a total of nearly 50,000 facts which were then available to disclose the performance of this experimental locomotive. It is perhaps safe to say that there was nowhere such an accumulation of locomotive data obtained

FIG. 22 —An Interior View, Second Plant.

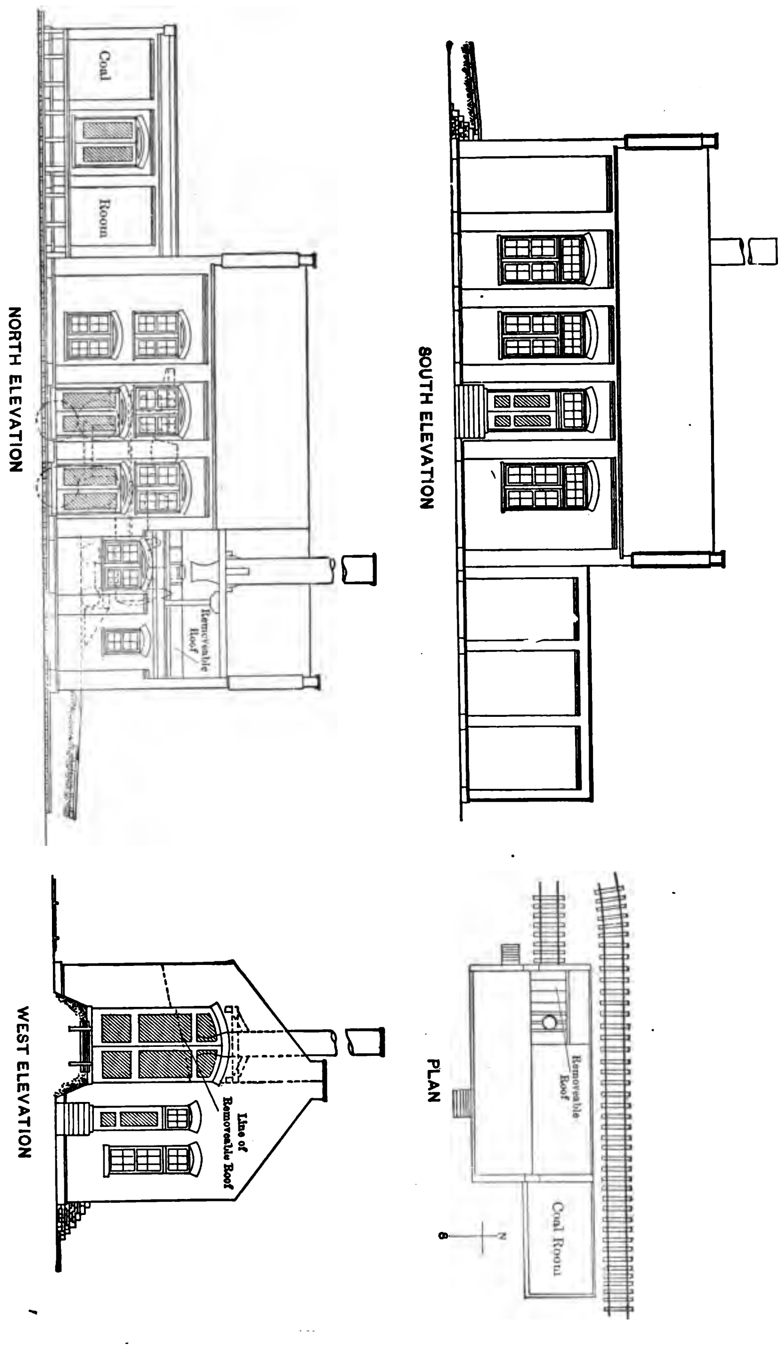

FIG. 23.—The Locomotive Laboratory, Purdue University, 1894.

FIG. 24.—The Locomotive Laboratory, 1894.

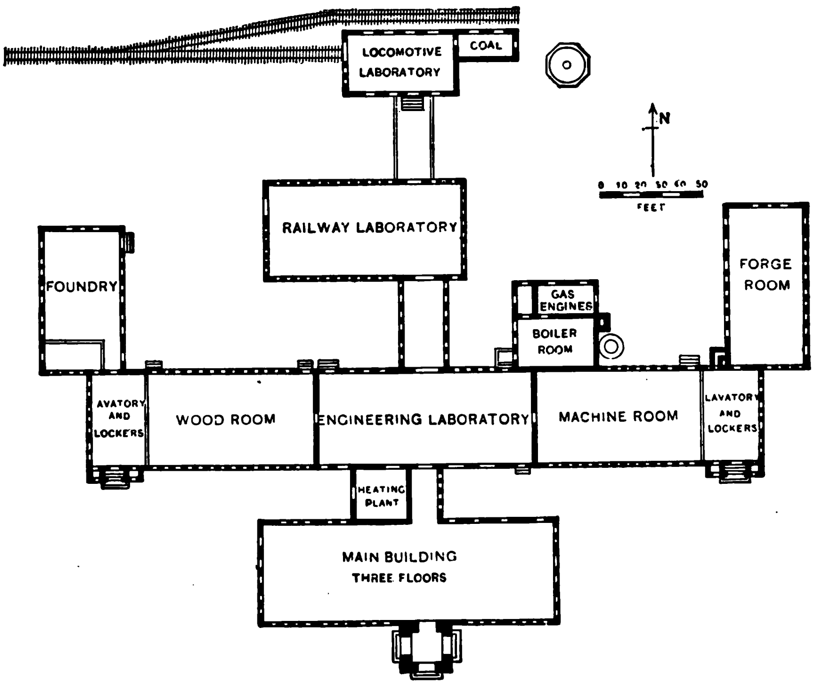

FIG. 25.—A Plan of the Reconstructed Engineering Laboratory, Purdue University, 1898.

under conditions so favorable to experimentation, and arranged so systematically, as the material then existing in the Purdue laboratory.

FIG. 26.—The Departure of Schenectady No. 1.

FIG. 27.—Schenectady No. 1 in a New Rôle.

16. Sale of Locomotive "Schenectady."—The six years which followed the introduction of Schenectady into the Purdue laboratory

had been marked by unusual progress in locomotive design, and by the end of that period the experimental engine had ceased to be representative of the most approved practice. For example, steam pressures of 180 or 200 pounds per square inch were then common, while the boiler of Schenectady had been designed for a maximum pressure of 140 pounds. Early in 1897, therefore, it was decided that locomotive Schenectady should be disposed of and another engine which would better serve the purposes of the

Fig. 28.—The Second Experimental Locomotive Schenectady No. 2, 1897.

laboratory secured to take its place. It thus happened that on a bright morning in May, 1897, Schenectady, hereafter to be known as Schenectady No. 1, steamed out of its "rectangular round-house" and slowly made its way across the campus under the escort of a throng of students. At the Art Hall a stop was made and the significance of the event was emphasized by brief addresses.*

Passing from Purdue's possession, Schenectady became the property of the Schenectady Locomotive Works, and later entered regular service on the Michigan Central Railway. Our last glimpse

* The speakers were President Smart, Colonel H. G. Prout, and two members of the faculty.

of the machine shows it at the head of a train on this road, where its identity is established by its number, 422.*

17. **Schenectady No. 2** arrived at the laboratory, fresh from its builder's hands, in October, 1897, and entered at once upon the work for which it was designed. Since, however, the greater part of the data discussed in succeeding chapters is the result of the work done on Schenectady No. 1, a description of Schenectady No. 2 will for the present be omitted.

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* The photograph from which Fig. 27 has been made was obtained through the courtesy of the late Mr. Robert Miller, Superintendent of Motive Power and Equipment of the Michigan Central Railway. In transmitting the photograph Mr. Miller wrote, under date of Dec. 11, 1897: "I take pleasure in forwarding you to-day . . . a photograph of our engine, No. 422, known to us as the Purdue Locomotive. Our men call her 'the schoolmarm.'"

CHAPTER II.

GROWTH OF INTEREST IN LABORATORY TESTS OF LOCOMOTIVES.

18. Locomotive Operation under Conditions other than those of the Track.—A testing-plant was not required to demonstrate the practicability of operating a locomotive with the machine as a whole at rest, for, previous to the existence of such a plant, locomotives had been made to serve the purposes of stationary engines by being blocked up until their drivers cleared the rails enough to carry belts; * and before the days of the injector it sometimes happened that an engine held for an unusual period on a siding was in an emergency pumped without changing its position. This was accomplished by setting a jack under the rear of the engine in such a manner as to take a considerable portion of the weight of the engine from the drivers, which, thus relieved of pressure and aided by the application of oil to the rail, could be slipped at moderate speed through the action of its cylinder while the pumps delivered feed-water to the boiler. Nor has the student of locomotive performance failed to take advantage of such possibilities as these. Thus, the late Alexander Borodin, when Engineer-in-Chief of the Russian Southwestern Railway, presented the results of an elaborate series of tests made upon a small locomotive blocked clear of the track, with belts applied to the drivers in such a way that the power developed could be absorbed in driving shop machinery.† But in neither of these cases was it possible to operate the locomotive under conditions normal to the track. The Purdue plant was the first to receive a normal engine in the condition in which it might be as it

* A locomotive thus arranged was seen driving the machinery of a railway repair-shop in Montgomery, Ala., in the winter of 1866.

† "Experiments on the Steam-jacketing and Compounding of Locomotives in Russia," by Alexander Borodin, Mem. Inst. M. E. Proceedings of the Meetings of the Institution of Mechanical Engineers, London, August, 1886.

came from the road, and to allow such an engine to be loaded and run in a perfectly normal way.*

19. Growth of Interest in Locomotive-testing.—The year 1891, which marked the installation of the Purdue locomotive testing-plant, and those immediately succeeding, represent a period of unusual interest in locomotive design and performance. The preceding twenty-five years had witnessed the development of the vast railway systems of this country, new engines had been constantly in demand for the operation of new track, and shop facilities for repairing them were required to be constantly increased. When there was an end to the period of enormous track extension, the conditions affecting locomotive development became more settled; railway managers began to enforce economy in operation, and attention was thus turned from matters of construction to questions of performance. As a consequence, much activity was developed in testing locomotives while in operation on the road, but there was no standard method of making such tests; and as each investigator followed his own method, a comparison of results proved to be of little value. These conditions led naturally to a desire for greater uniformity in practice, which, in the spring of 1890, found expression at a meeting of the American Society of Mechanical Engineers, which resolved "That a Committee of Seven be appointed by the Chair to report on Standard Methods of Conducting Tests of Efficiency of Locomotives, including the engine, the boiler, the quality of the steam, and the comparative efficiencies of simple and compound locomotives." †

The debate upon the resolution there adopted revealed the limited and indefinite character of the information available concerning the performance of locomotives, and gave expression to a desire for fuller knowledge upon the subject.

In the early summer of the following year it happened that the American Society of Mechanical Engineers and the American Railway Master Mechanics' Association met during the same week (June,

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* In this connection, it will be well to note that in the course of a discussion before the American Society of Mechanical Engineers in May, 1890 (Proceedings of the Society, Vol. XI, p. 886), Mr. Louis S. Wright proposed a scheme for testing locomotives embracing many of the important features which appear in the Purdue plant, as designed a few months later, and it is probable that the importance of the problem had led others to consider it as a subject for speculation. It is but proper to add that the Purdue designer did not draw his inspiration from any such suggestion.

† Proceedings of the American Society of Mechanical Engineers, 1890, p. 591.

1891), one at Providence, R. I., and the other at Cape May, and it happened also that the subject of locomotive testing was introduced in both conventions. The Mechanical Engineers received a report of progress * from its committee appointed a year previous, and the Master Mechanics appointed a committee "to investigate the practicability of establishing a standard system of tests to demonstrate the fuel and water consumption of locomotives; also to ascertain the value of the steam-engine indicator for locomotive use." † In the succeeding year (June, 1891, to June, 1892) the Mechanical Engineers' Committee made no report, but in December, 1892, it again reported progress.‡ During this interval the Master Mechanics' Committee had asked and received authority to confer with the Committee of the Mechanical Engineers, but the official records show no further reference to the work of either committee or to any joint work done by them until the summer of 1893, when the Committee of the Master Mechanics' Association presented their report after conference with a similar committee appointed by the American Society of Mechanical Engineers.§ The report gave in elaborate form the specifications to be observed in conducting tests of locomotives upon the road. It is an interesting fact that up to this time the record contains no reference to laboratory tests or to locomotive-testing plants.

20. Interest in Purdue's Work.—A paper describing Purdue's locomotive-testing plant was presented at the San Francisco meeting of the American Society of Mechanical Engineers in the summer of 1892,‖ where it failed to draw out any discussion. But a year later, when the Master Mechanics' Committee already referred to rendered its report, the Convention after due discussion resolved, first, "That the Committee on Locomotive Tests be instructed to cooperate with the Committee from the Mechanical Engineers' Society in a series of comparative shop tests of compound and simple locomotives to be made on the locomotive-testing apparatus of Purdue University," and, second, "That the Executive Committee be instructed

* Proceedings of the American Society of Mechanical Engineers, 1891, p. 613.

† The committee appointed in response to this resolution consisted of J. N. Lander, J. Davis Barnett, Albert Griggs, John D. Campbell and F. W. Dean. (Proceedings of the American Railway Master Mechanics' Association, 1891, p. 206.)

‡ Proceedings of the Society, 1893, p. 21.

§ Proceedings of the American Railway Master Mechanics' Association, 1893, p. 22.

‖ "An Experimental Locomotive." Transactions of the American Society of Mechanical Engineers, Vol. XIII, p. 427.

to provide funds to pay the necessary expenses connected with such shop tests, and solicit subscriptions on this account from railroad companies and locomotive-builders."

A few weeks later, at the summer meeting of the American Society of Mechanical Engineers, a paper giving results of twenty efficiency tests upon the Purdue locomotive was presented,* and at this meeting, also, the committee appointed to report upon standard methods of conducting locomotive tests presented a document † dealing in a comprehensive manner with the question at issue, giving liberal attention to shop tests and recommending such tests as the only satisfactory method of settling the important questions of performance. As a result of this report a resolution was proposed calling for a committee to cooperate with the committee of the American Railway Master Mechanics' Association in carrying out investigations at Purdue. At the next regular meeting of the Society, however, the council to whom the above-mentioned resolution was referred reported that such a resolution was deemed inexpedient, and further consideration of the matter by the Mechanical Engineers was, therefore, dropped.

Meanwhile it was being urged by the Master Mechanics' Association that funds and locomotives should be made available for work upon the Purdue plant, and while the movement here was also destined to failure, so far as the immediate purpose in view was concerned, the discussion and reports during the years 1893 to1895 leave no doubt as to the attitude of the Association with reference to the value of laboratory tests of locomotives.

21. **New Plants.**‡—In view of the value of the locomotive-testing plant as a means of determining the performance of locomotives, it was reasonable to expect that the number of such plants would increase. The first to be completed, following lines developed at Purdue, was that of the Chicago & Northwestern Railroad Company, which went into commission in the year 1895; the next that of Columbia University, built in 1899. Following these came various plants for railroad companies and technical institutions both in this country and abroad, among which the most deserving of mention at this time (1906) is probably that of the Pennsylvania Railroad Company installed at St. Louis during the Exposition of 1904 and afterwards removed to the Company's shops at Altoona.

* "Tests of the Locomotive at the Laboratory of Purdue University." Transactions of the American Society of Mechanical Engineers, Vol. XIV, pp. 826–854.

† Transactions of the American Society of Mechanical Engineers, 1893, pp. 1312–1339.

‡ Transactions of the American Society of Mechanical Engineers, Vol. XXV.

CHAPTER III.

LOCOMOTIVE SCHENECTADY NO. 1.

22. **The Controlling Conditions Affecting the Choice of a Locomotive** which should serve the purposes of the laboratory were the outcome of a desire to have the machine, in size and in the character of its details, fairly representative of good American practice at the time the order was given; that is, in 1891. The type of locomotive and the size of its cylinders were the only points prescribed by the University authorities. The details of the design were left to

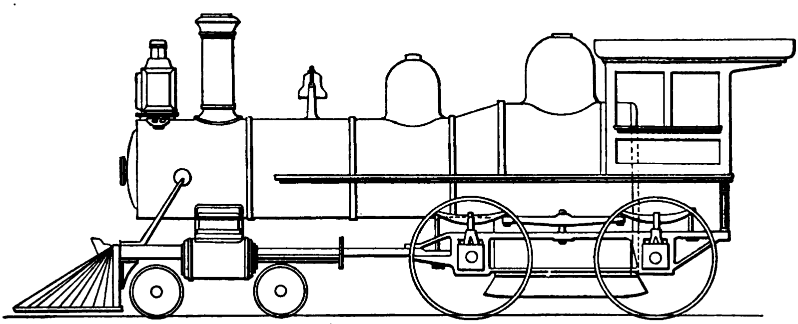


FIG. 29.—Elevation of Schenectady No. 1.

the discretion of the builders, who, being thus untrammeled, produced an engine which it is presumed represented their standard practice. There were heavier and more powerful engines in use, but there were also very many which were lighter and of less power. The locomotive was in every respect normal. No fixture or fitting was omitted which was then in common use on the road. It had air-brakes and a headlight, and had it been coupled to a tender and placed at the head of a train, it would have lacked nothing for the service which might then have been demanded of it. Neither

was there any detail of consequence added or modified by the builders to especially adapt the machine to the purposes of the laboratory.

The locomotive, therefore, being representative in size and normal in character, it was but reasonable to expect that results obtained from it would, in their general aspect at least, apply to a very large class of American locomotives.

23. **Specifications** as prepared by the builders are as follows:

SCHENECTADY LOCOMOTIVE WORKS.

SCHENECTADY, N. Y., April, 1891.

SPECIFICATION No. 1491.

Eight-wheel Locomotive Engine for Purdue University: Engine Schenectady, No. 1.

GENERAL DESCRIPTION.

Gauge 4 feet 8½ inches. Fuel, bituminous coal.
Cylinders 17″ diam., 24″ stroke. Drivers 63″ diam.
Driving-wheel base 8 ft. 6 in. Total wheel-base 22 ft. 11 in.
Weight in working order, on drivers, about 56,000 lbs.
Total, about 85,000 lbs.

CONSTRUCTION.

BOILER. To be of the best workmanship and material, to be capable of carrying with safety a working pressure of 140 lbs. per square inch, and of sufficient capacity to supply steam economically. All horizontal seams quadruple-riveted with welt-strip inside. A double-riveted seam uniting waist with fire-box. All plates planed at edges and calked with round-pointed tools. Boiler to have extended front end. Waist, dome, and outside of fire-box of steel $\frac{7}{16}$″ thick. Diam. of waist at front end 52″, made wagon-top, with one dome 30″ diam. placed on wagon-top.

FIRE-BOX. Of best quality fire-box steel, 72″ long, 34⅜″ wide, 73″ deep. Crown-sheet three-eighths, tube-sheet one-half, side and back sheets five-sixteenths thick. Water-space 4″ front, 3″ sides, 3″ back. All sheets thoroughly annealed after flanging.

Stay-bolts ⅞ and 1 inch diam., screwed and riveted to sheets, and placed not over 4 inches from center to center.

Crown-sheet supported by crown-bars made of two pieces of wrought iron 5″ wide, ¾″ thick, placed not over 4½ inches apart, reaching across crown and resting on edge of side-sheets. Crown-bars riveted to crown-sheets, with seven-eighth-inch rivets placed not over 4½ inches from center to center, each bar having four stay-braces to top of boiler or dome.

TUBES.	Of charcoal-iron, 200 in number, 2″ diam., 11 ft. 6 in. in length. Set with copper ferules at both ends. Cleaning-holes at the corners of fire-box, and blow-off cock on back.
GRATE.	Grates, rocking. Ash-pan with dampers front and back.
STACK.	Smoke-stack straight. Deflecting-plate and netting in smoke-box.
FRAMES.	Of best hammered iron, main frame in one section with braces welded in. Forward section securely bolted and keyed to main frame. Pedestals protected from wear by cast-iron shoes and wedges, and locked together at bottom by bolt through cast-iron thimble. Width of main frame 4″.
CYLINDERS.	Of close-grained hard charcoal-iron. Cast with half-saddle attached, the right and left cylinders from the same pattern and interchangeable. Fitted together in a substantial manner and securely bolted and keyed to frame. Valve-face and steam-chest seat raised above face of cylinder to allow for wear. Cylinders oiled from N. & Co. No. 8 sight-feed lubricator placed in cab, with copper pipe under boiler-lagging to steam-chest.
THROTTLE.	Balanced valve placed in dome, with wrought-iron dry-pipe and cast-iron steam-pipe connecting to cylinders.
PISTONS.	Made with removable followers, or with solid heads, and fitted with approved steam-packing. Piston-rods of hammered iron.
VALVE-MOTION.	Approved shifting link-motion graduated to cut off equally at all points of the stroke. Links, sliding-blocks, plates, lifting-links, pins, and eccentric-rod jaws of the best hammered iron, thoroughly case-hardened. Steam-chest valves, Richardson balanced.
GUIDES.	Of hammered iron, case-hardened.
CROSS-HEADS.	Of cast steel with brass gibs. Style for four bar-guides.
DRIVING-WHEELS.	Four in number, 63″ diam. Centers cast of the best charcoal-iron and turned to 57″ diam. to receive tire.
TIRES.	Of American steel, 3″ thick, both pairs flanged $5\frac{3}{4}$″ wide.
AXLES.	Of hammered iron, with journals 7″ diam. 8″ long. Driving-boxes of cast iron, with heavy Damascus bronze bearings and large oil-cellars.
SPRINGS.	Made of the best cast steel, tempered in oil. Secured to a system of equalizing-beams to insure the engine riding in the best possible manner.
RODS.	Connecting and parallel rods of hammered iron or steel, each forged solid, fitted with all necessary straps, keys, bolts, and brass bearings.

CRANK-PINS.	Of steel.
WATER-SUPPLY.	To have two injectors (Monitor No. 8 R. H., No. 7 L. H.) with well-arranged cocks and valves for convenience in working.
ENGINE-TRUCK.	With square wrought-iron frame, cast-iron pedestals, and center bearing suitable for rigid center, with approved arrangement of equalizing beams and springs.
WHEELS.	Four double-plate chilled wheels of first-class manufacture, 28″ diameter.
AXLES.	Of hammered iron, with inside journals 5″ diam., 9″ long. Springs of best cast steel, tempered in oil.
CAB.	Constructed of seasoned ash, substantially built, and secured by joint-bolts and corner-plates. Furnished with seats and tool-boxes for engineer and fireman.
PILOT.	Of wood, strongly braced, and provided with substantial draw-bar.
FINISH.	Boiler lagged with wood and jacketed with planished iron, secured by planished iron bands. Dome lagged with wood, with sheet-iron casing and cast-iron rings painted. Cylinders lagged with wood, with sheet-iron casing and cast-iron head-covers finished. Steam-chests cased with sheet iron, with cast-iron covers finished. Hand-rail of iron finished, with columns secured to boiler. Boiler front and door of cast iron.
FIXTURES AND FURNITURE.	Engine provided with sand-box, support for headlight, bell, whistle, steam-gauge, gauge-cocks, cab-lamp, blower, oil-cans, torch; also all necessary wrenches, fire-tools, hammers, chisels, packing-tools, etc. Two jack-screws and a pinch-bar. Principal parts of engine fitted to gauges and templates, and interchangeable. All finished removable nuts case-hardened. I. A. Williams & Co. headlight. All threads United States standard.
PAINTING.	Engine to be well painted and varnished, with the road, number mark, and name put on in handsome style.
PATENTS.	All patent fees not covered by this specification excepted.
BRAKE.	Westinghouse automatic air-brake on drivers.

24. Drawings of the locomotive as covered by the preceding specifications follow the text of this chapter.

25. Constants.—In calculating the performance of the locomotive from observed data, use is made of various dimensions, relationships,

and calculated values, which for convenience are herein grouped under a single head as "constants." The dimensions upon which some of these values are based, as, for example, cylinder diameters, are subject to slight change resulting from use or repairs; and as the tests in question extended over a period of several years, and as they were interrupted by the necessity of giving the locomotive general repairs, the values used as constants were carefully watched and frequently checked. All changes observed were, however, too small to merit consideration excepting those which occurred in the diameter of the cylinder at the time the engine was put through general repairs in 1894. After this date certain values which had previously prevailed could no longer be employed, and it will be noted that the tabulated statement exhibits two sets of constants for such factors.

At the time these tests were made, authorities disagreed as to the method of measuring the extent of heating surface in boilers, some advocating the use of the inner surface, while others favored the use of the outer surface of the tubes. The general practice in locomotive work had been to employ the outside of the tube. In the Purdue work, for reasons which appeared logical *the heating surface was defined as the inside surface of the fire-box plus the inside surface of the tubes plus the effective surface of the front head.*

A summary of the principal dimensions of the experimental locomotive and of constants used in calculating results derived from tests is as follows:

Total weight (makers' figures)	85,000 pounds.
Weight on four drivers (makers' figures)	56,000 "
Total wheel-base	22 ft. 11 in.
Driving-wheel base	6 " 6 "
Nominal cylinder diameter	17 in.
Nominal stroke of piston	24 "
Diameter of piston-rods	3 "
Area of piston-rod section	7.07 sq. in.

	Prior to Jan. 24, 1894.	Subsequent to Jan. 24, 1894.
Actual cylinder diameter (inches):		
Right side	17	17.047
Left side	17	17.031
Actual stroke of pistons (inches):		
Right side	24	23.936
Left side	24	23.889

		Prior to Jan. 24, 1894.	Subsequent to Jan. 24, 1894.
Effective area of pistons (square inches):			
Right side	Head end	226.98	228.23
	Crank end	219.91	221.16
Left side	Head end	226.98	227.82
	Crank end	219.91	220.75
Piston displacement, cubic feet:			
Right side	Head end	3.1525	3.1617
	Crank end	3.0540	3.0637
Left side	Head end	3.1525	3.1497
	Crank end	3.0540	3.1520
Clearance volume, per cent of piston displacement:			
Right side	Head end	9.390	9.103
	Crank end	10.265	10.046
Left side	Head end	9.980	10.086
	Crank end	10.019	9.880
INDICATED H.P. CONSTANT, or the horse-power for one pound mean effective pressure and a speed of one revolution per minute.			
Right side	Head end	.01375	.013796
	Crank end	.01332	.013368
Left side	Head end	.01375	.013744
	Crank end	.01332	.013318
ractive H.P. Constant, or the horse-power developed at the draw-bar when there is a pull of one pound and a speed of one revolution per minute		.0004981	.0004956
Drivers:			
Nominal diameter, inches		63	63
Actual circumference, inches		197.25	196.25
Actual diameter, inches		62.7	62.47
Ports:			
Length, inches		16	
Width of steam-ports, inches		$1\frac{1}{4}$	
Width of exhaust-port, inches		$2\frac{1}{2}$	

Valves:

Type—"D" slide.

Design—Richardson balanced.

The proportions and the setting of the valves were changed for different series of tests as follows:

Reference Symbols.		Outside Lap. Inches.	Inside Lap. Inches.	Inside Clearance. Inches.	Valve-setting.
No. appearing in Col. 5, Table I.	Series.				
I	V	$\frac{3}{4}$	$\frac{1}{32}$	..	Builders'
II	A	$\frac{3}{4}$	$\frac{1}{32}$	..	Lead Reduced
III	B	$\frac{3}{4}$	$\frac{1}{32}$	..	
IV	C	$\frac{3}{4}$	$\frac{1}{32}$	..	
V	E	$\frac{3}{4}$	$\frac{1}{32}$	..	
VI	F	$\frac{3}{4}$	$\frac{1}{32}$	..	
VII	G	$\frac{3}{4}$	..	$\frac{1}{16}$	
VIII	H	$\frac{3}{4}$	..	$\frac{3}{16}$	
IX	I	$\frac{3}{4}$	..	$\frac{3}{8}$	
X	J	$\frac{3}{4}$	$\frac{1}{32}$	..	
XI	K	1	$\frac{1}{32}$	..	Eccentrics Advanced

Boiler:

Diameter waist at front end, inches	52
Diameter tubes, outside, inches	2
Diameter tubes, inside, inches	1.78
Number of tubes	200
Length of tubes, feet	11.5
Thickness of tubes, inches	.11
Area of flameway through tubes, feet	3.46
Width of fire-box, inches	34.5
Length of fire-box, inches	72
Height of fire-box, inches	73
Heating surface, square feet:	
In fire-box	132.1
In front head	10.5
In tubes calculated from outside diameter	1204.3
In tubes calculated from inside diameter	1071.8
Total heating surface assuming the tube surface to be calculated from the outside diameter	1346.9
Total heating surface assuming the tube surface to be calculated from the inside diameter.*	1214.4
Grate area, square feet	17.25
Ratio of heating surface to grate area:	
Assuming heating surface to be 1214.4 sq. ft.	7.04
Pounds of water in boiler when filled to middle gauge	8450
Steam-space in boiler when filled to middle gauge, cubic feet	52.8
Ratio of steam-space to entire cubical capacity of boiler	.29

Exhaust nozzle, double.

* This factor has been used in all calculations involving heating surface.

Weight of parts connecting with crank-pins:

Piston and piston-rod, pounds	297.0
Cross-head with part of indicator rigging attached, pounds	170.5
Main rod, pounds	344.5
Side rod, pounds	278.0

26. Steam-passages.—The areas of the several steam-passages between the throttle and the exhaust-tips through which the steam passes are shown graphically with great clearness in the diagram Fig. 54. By use of this diagram in connection with the recorded data of tests it should be possible to make an approximate determination of the steam velocity at various points in its course.

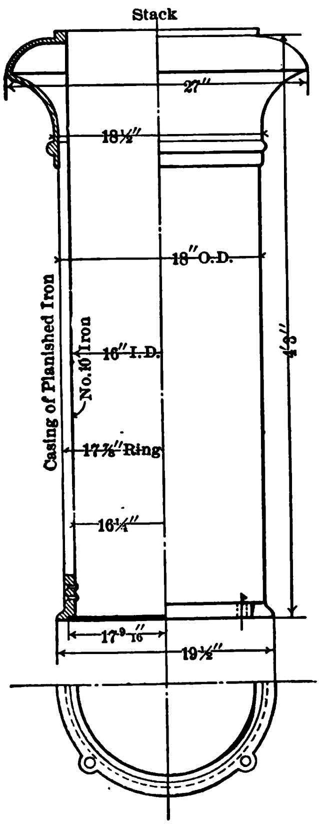


FIG. 30.—Stack.

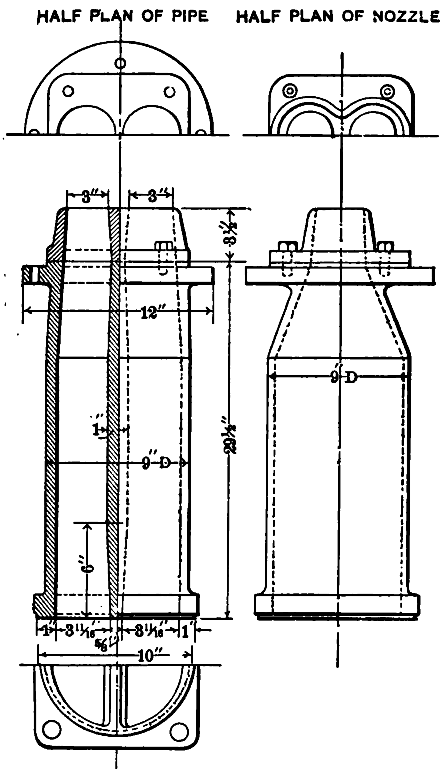


FIG. 31.—Exhaust-pipe and -nozzle.

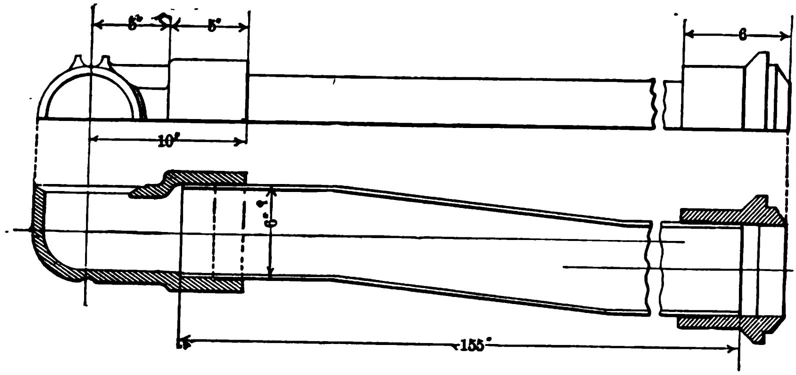


FIG. 32.—Dry-pipe.

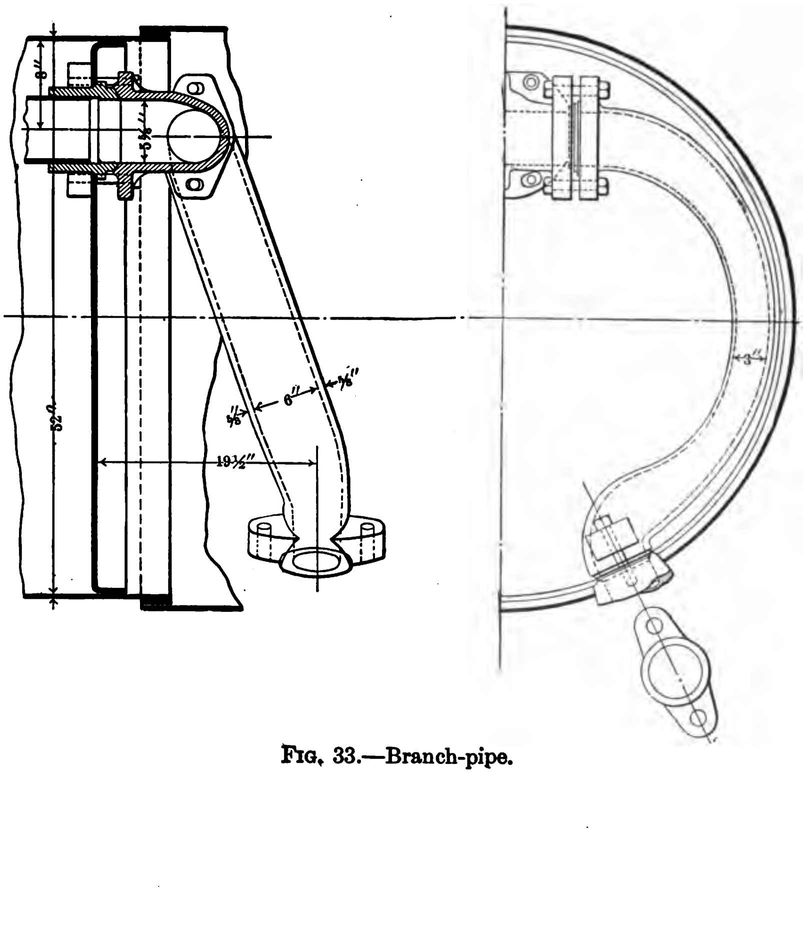


FIG. 33.—Branch-pipe.

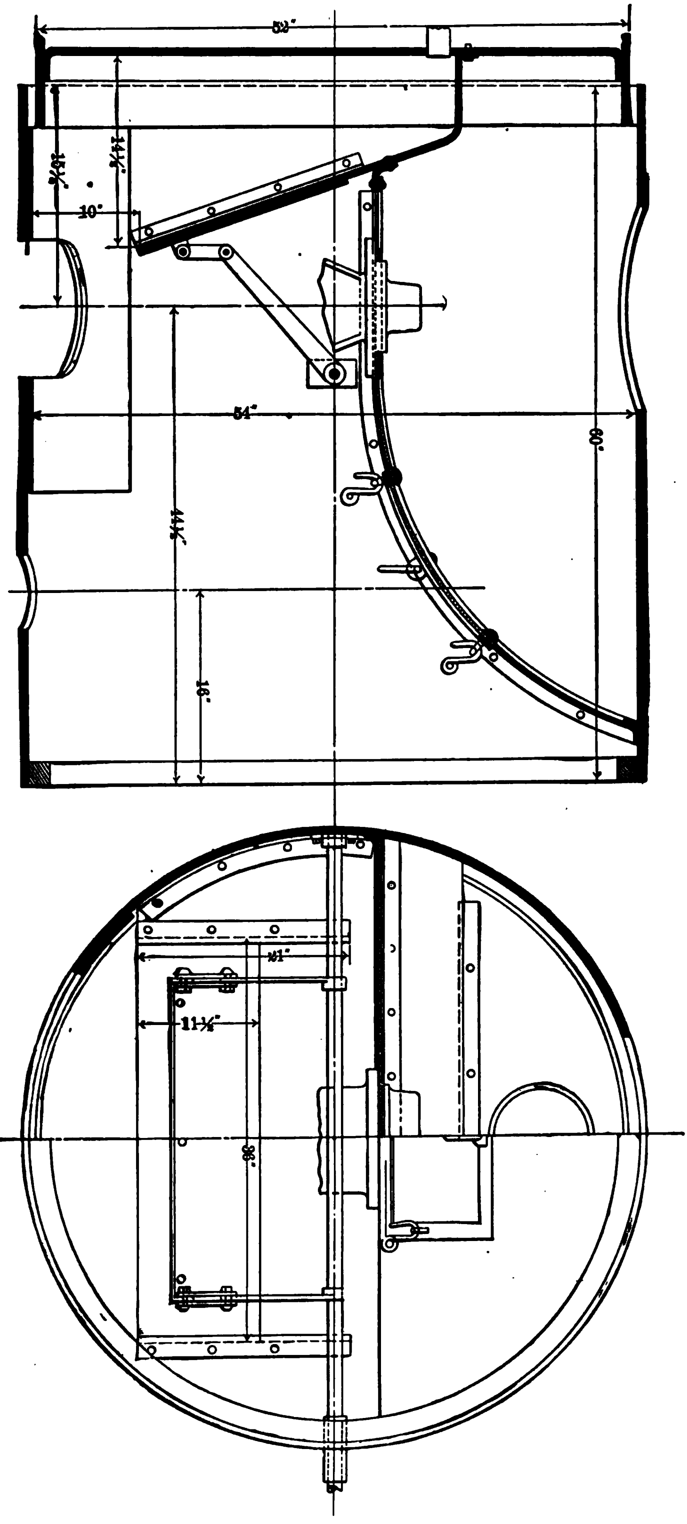


FIG. 34.—Netting and Deflector-plate.

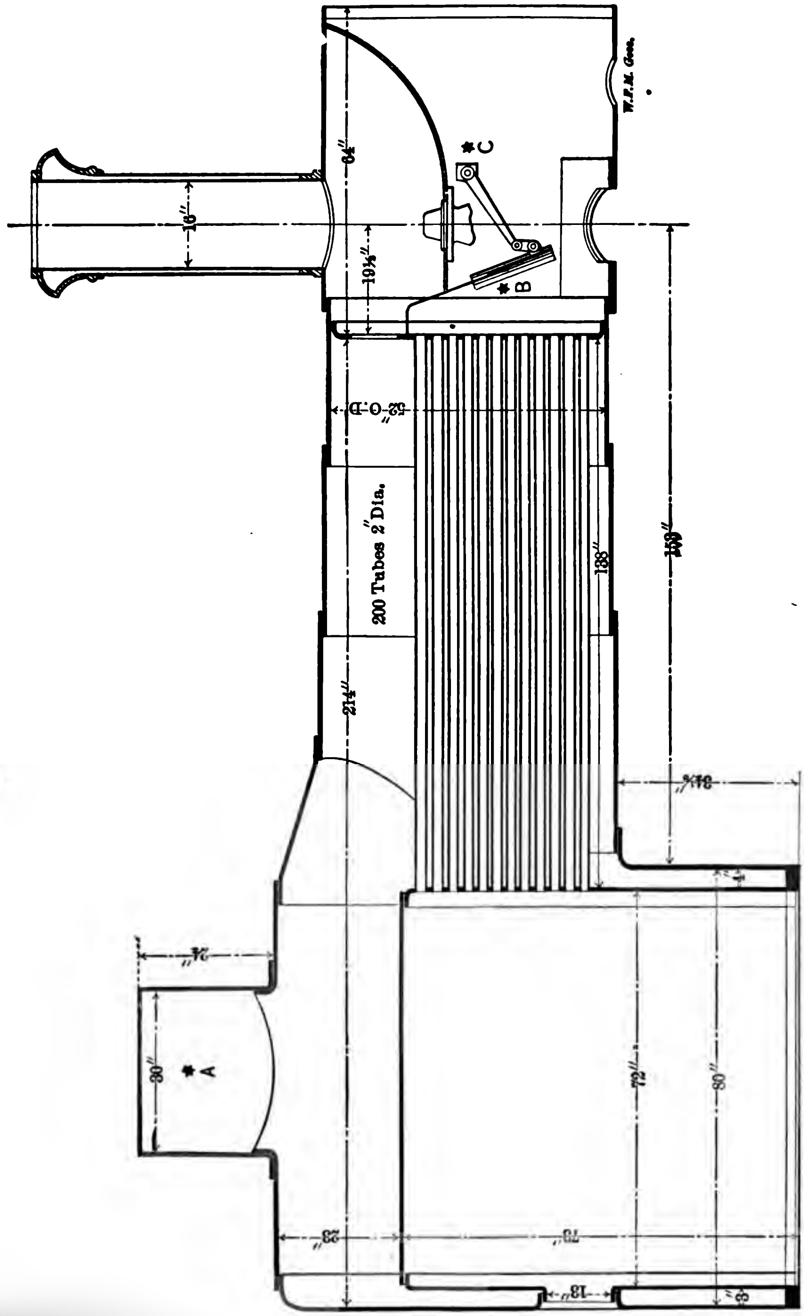


FIG. 35.—Boiler.

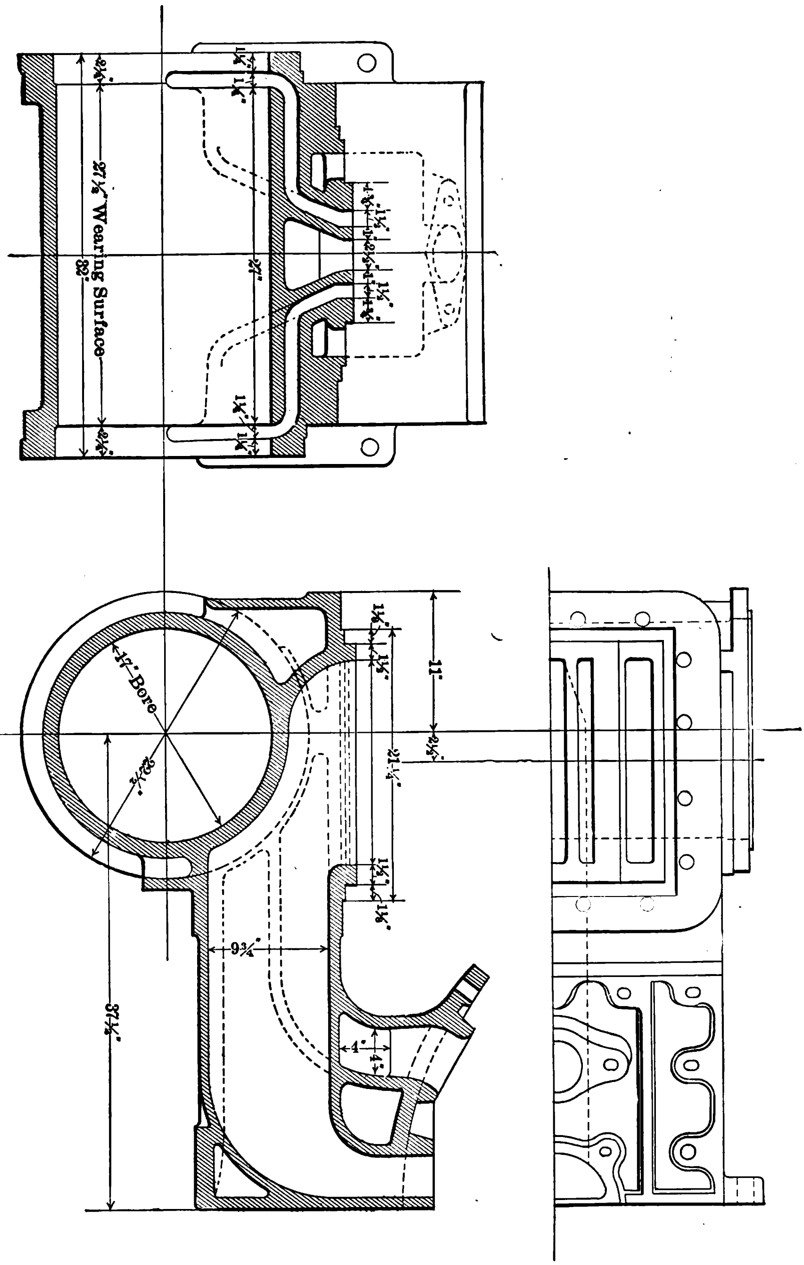


FIG. 36.—Cylinder and Saddle.

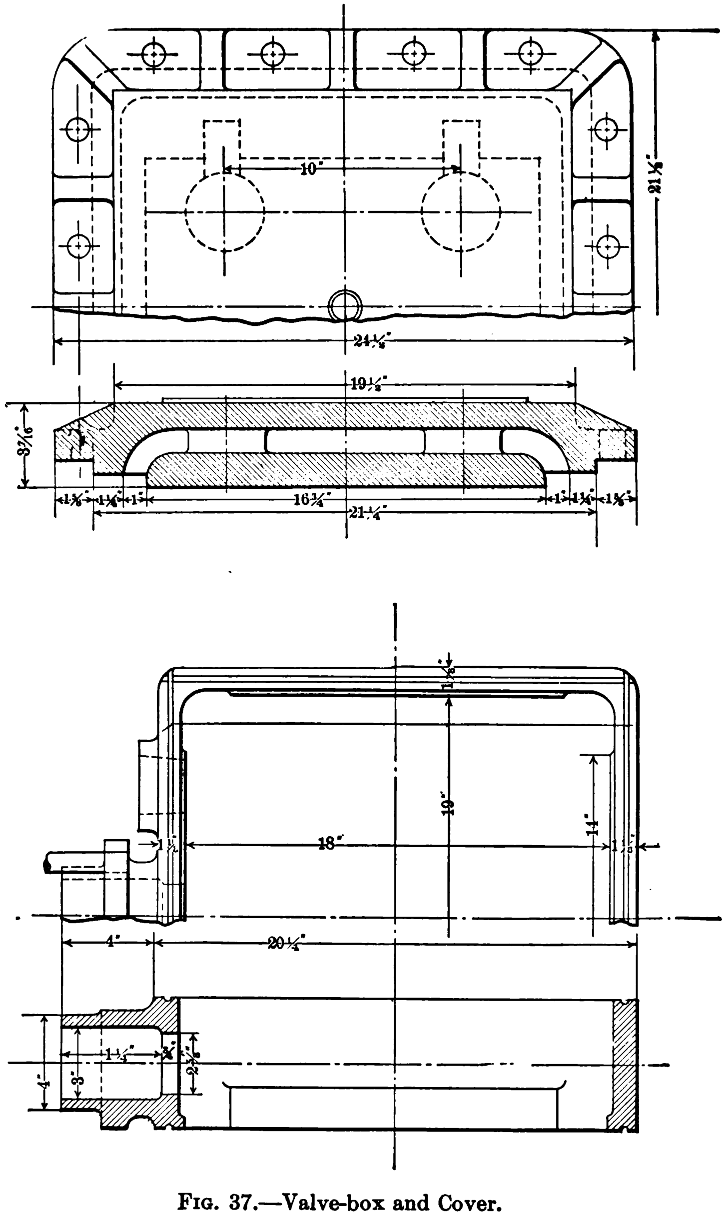


FIG. 37.—Valve-box and Cover.

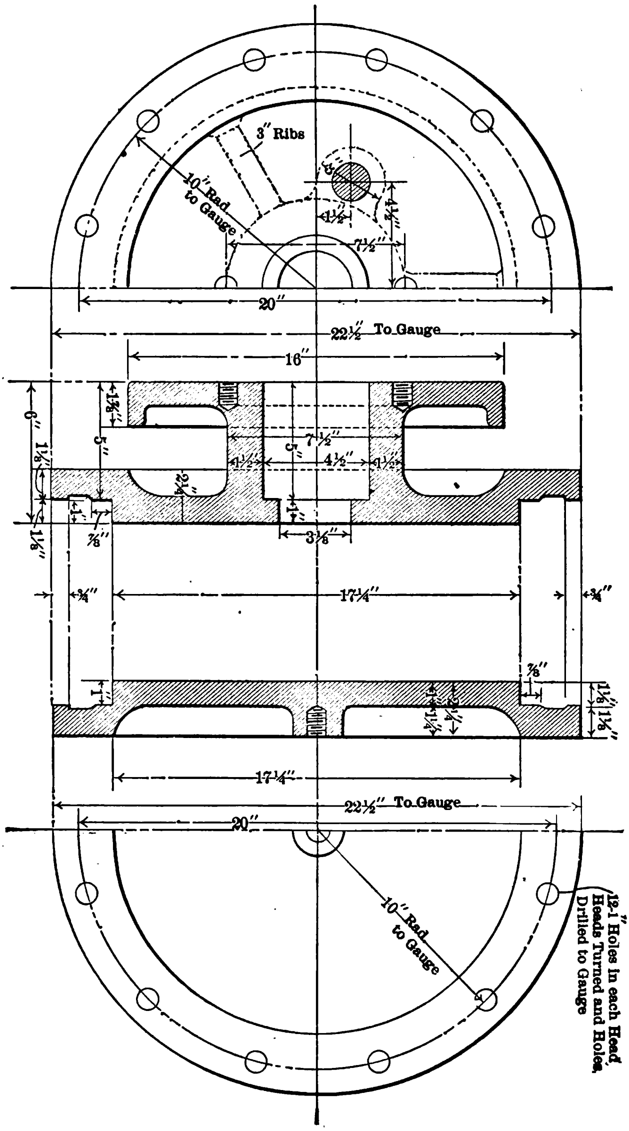


FIG. 38.—Cylinder-heads.

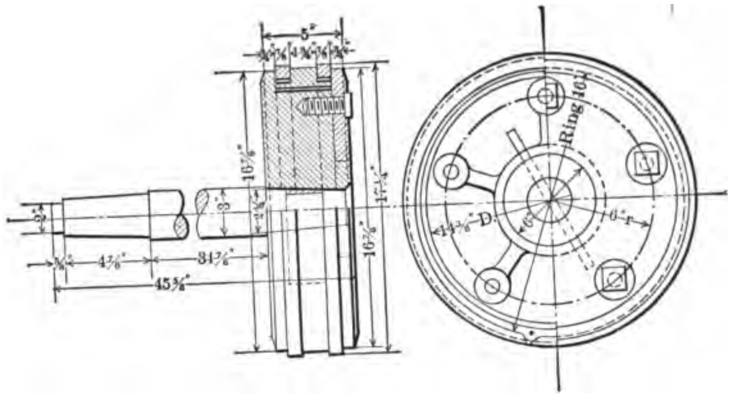


FIG. 39.—Piston and Piston-rod.

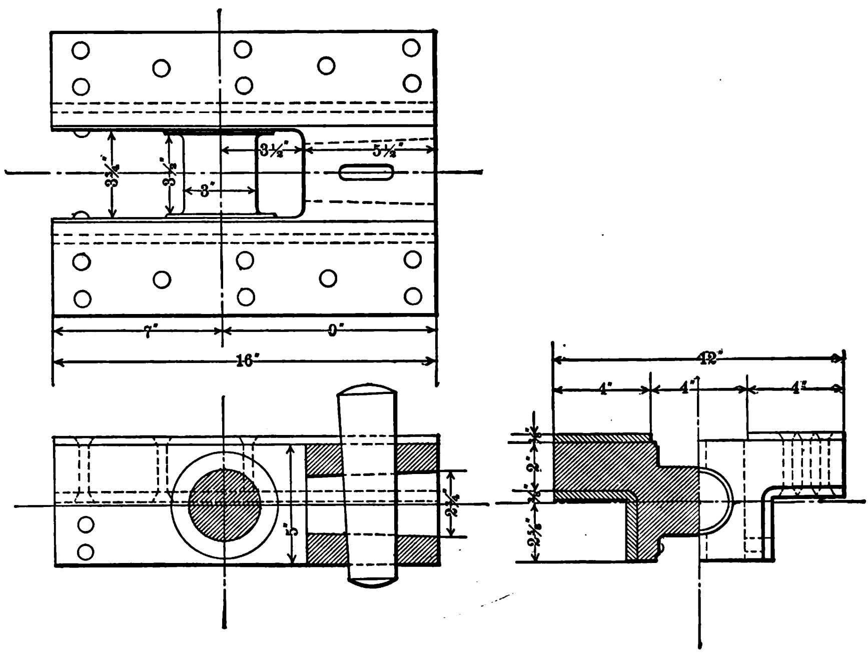


FIG. 40.—Cross-head.

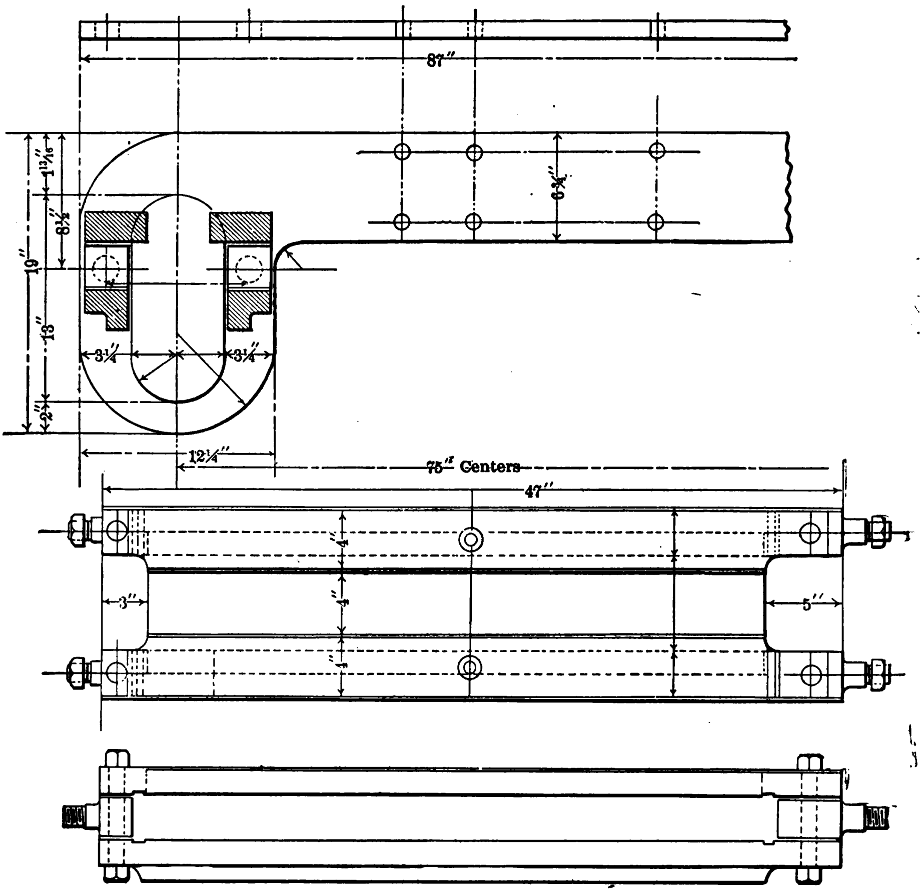


FIG. 41.—Guides and Guide-yoke.

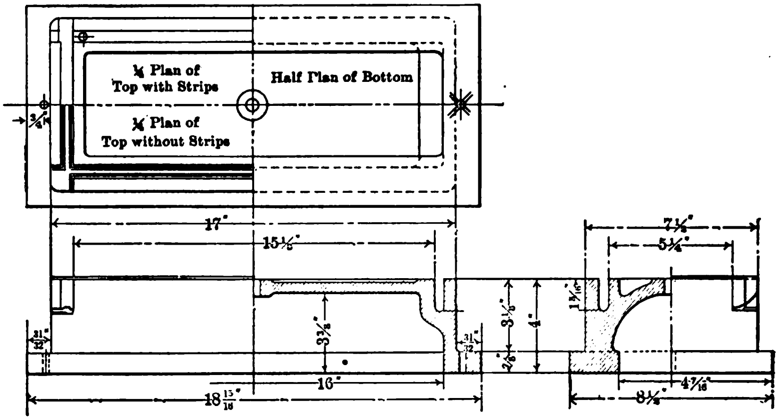


FIG. 42.—Valve.

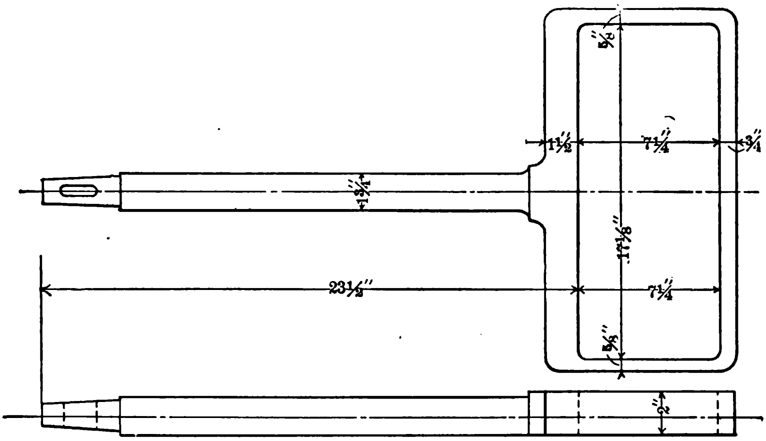


FIG. 43.—Valve-yoke.

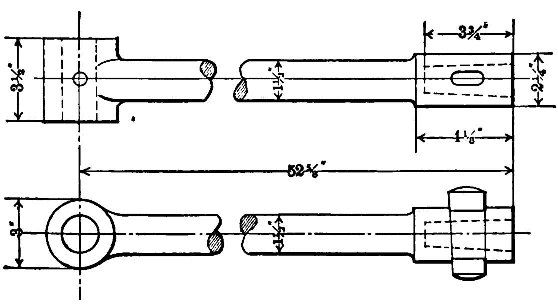


FIG. 44.—Steam-chest Valve-rod.

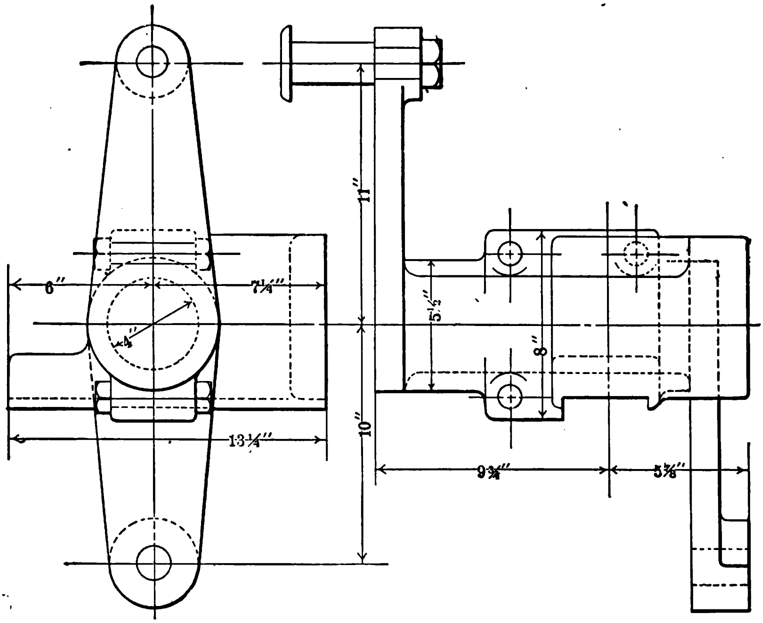


FIG. 45.—Rocker.

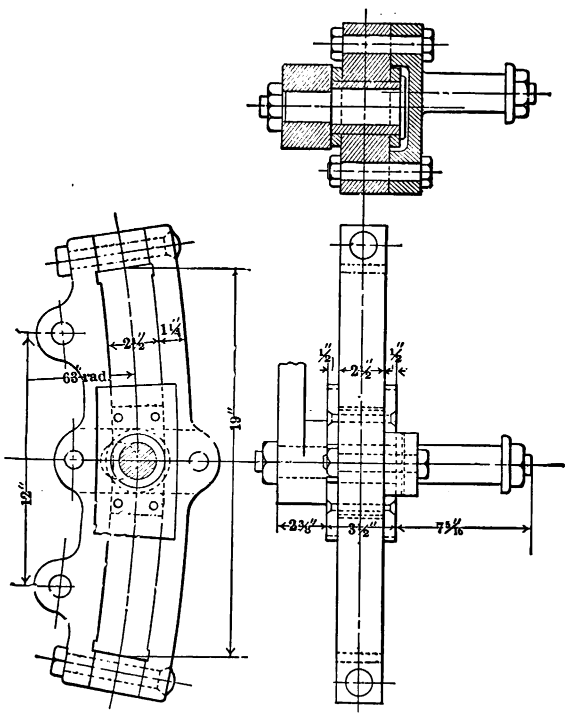


FIG. 46.—Link and Link-block.

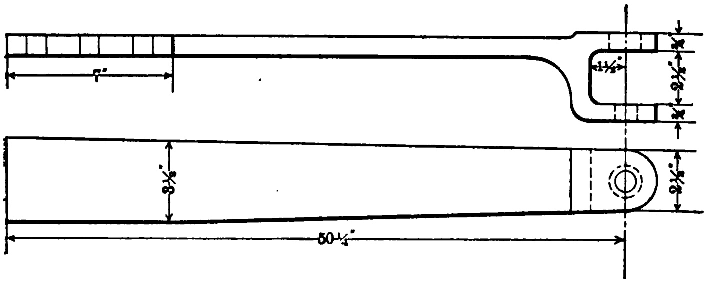


FIG. 47.—Eccentric Rod.

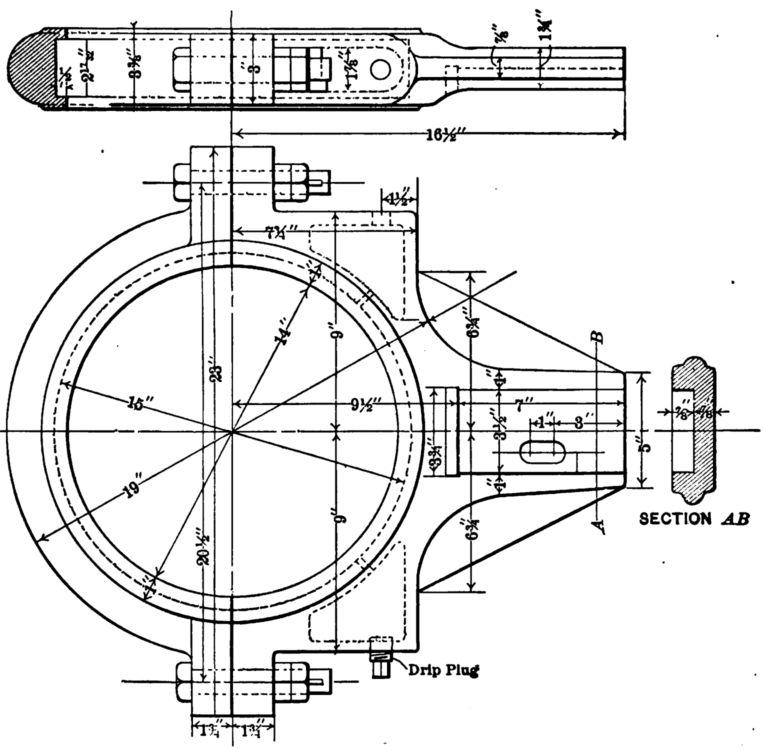


FIG. 48.—Eccentric Strap.

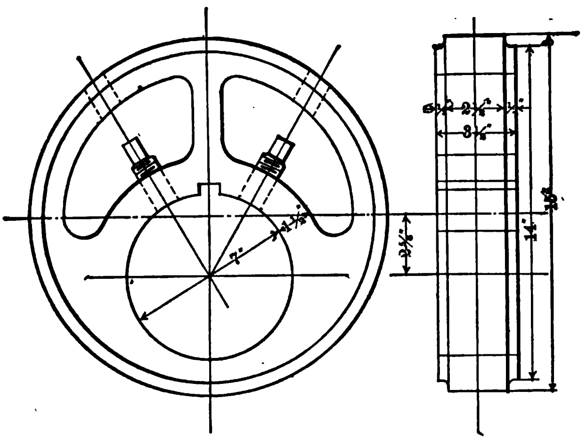


FIG. 49.—Eccentric.

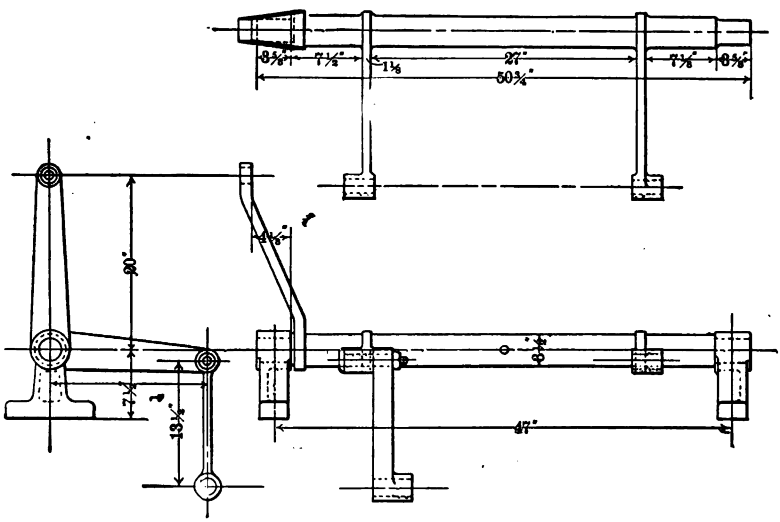
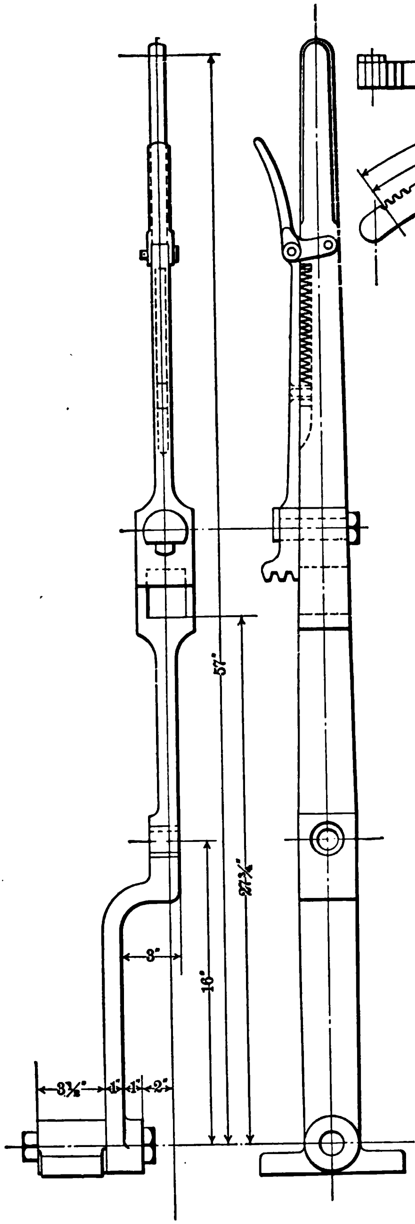


FIG. 50.—Reverse-shaft.



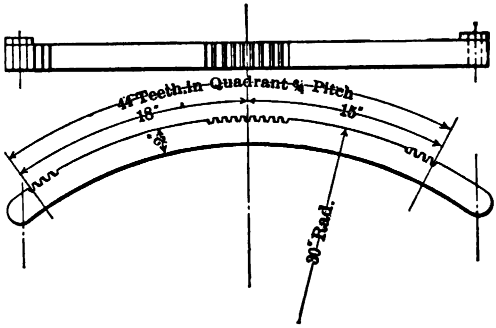


Fig. 51.—Reverse-lever and Quadrant.

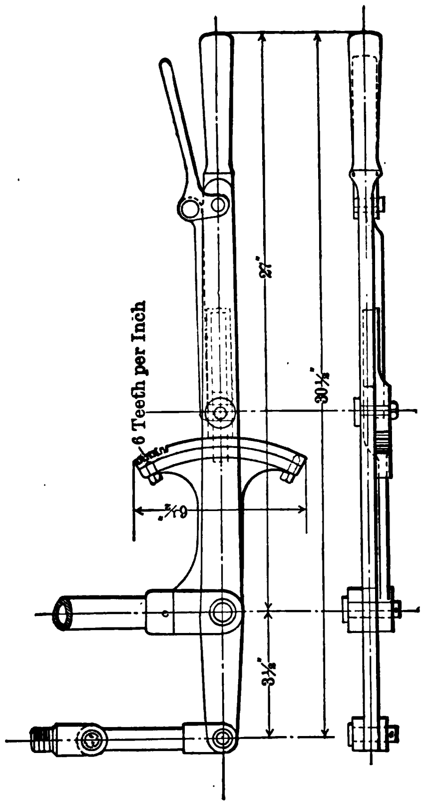


Fig. 52.—Throttle-lever.

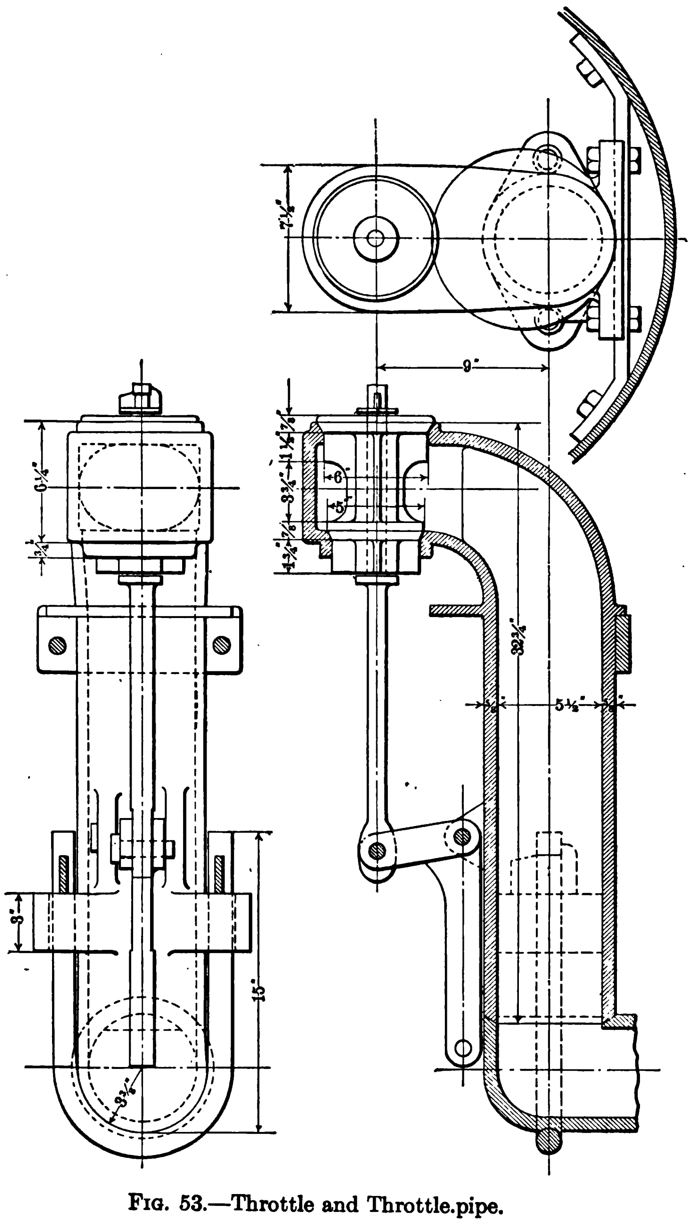


FIG. 53.—Throttle and Throttle.pipe.

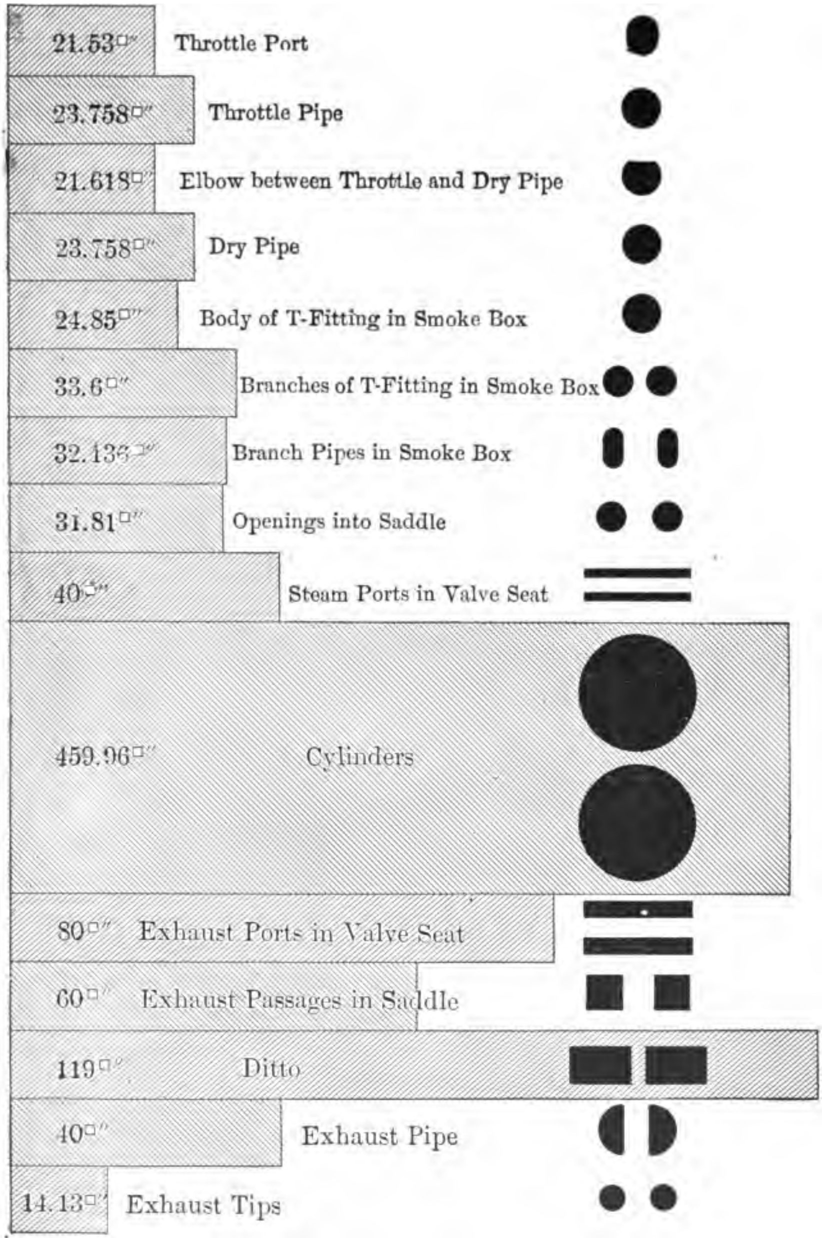


Fig. 54.—Steam-passage Areas.

CHAPTER IV.

METHOD OF TESTING AND DATA.

27. **Method of Testing.**—In preparation for a test the locomotive was run for an hour or two in the morning at low speed and light load. After the noon stop it was again started with the reverse lever in the position which it was to occupy during the test, while the load was gradually thrown on and the speed increased until the predetermined conditions for the test had been attained. At the stroke of a gong, usually between ten and thirty minutes after starting, all readings were taken and the test was commenced. The throttle was seldom touched after starting a test. Variations in boiler pressure during a test were so slight that changes in speed caused thereby were easily controlled by increasing or decreasing the brake load. At the end of the test care was taken to have the water in the boiler at the same level and the fire in the same conditions as at the start. Then upon the stroke of the gong the throttle was closed and the fire immediately dumped and quenched. Four regular attendants of the laboratory, including a fireman, prepared the locomotive for test, and during the tests fired the boiler and gave all necessary attention to the mounting and accessory machinery. A corps of student observers working under the direction of an instructor acting as supervisor of tests was responsible for all the observations. Students thus employed were generally seniors in the Department of Mechanical Engineering. Twelve observers were required, stationed as follows:

	Number of Observers.
To keep running log, time, and gong.	1
To weigh fuel; also to note time of starting and stopping injector, and to keep log of same.	2
To weigh feed-water, to observe temperature, and to keep log of same	2
To operate indicators; also to take calorimeter readings, dry-pipe pressure, draft-gauge, and pyrometer readings.	4
To read engine and supporting-wheel counters and dynamometer; also to read boiler pressure, and regulate the water pressure upon the several brakes.	2
To occupy the engineer's box.	1
	12

ENGINEERING LABORATORY. PURDUE UNIVERSITY.

RUNNING LOG

Of Test No. 95-1 Series A of LOCOMOTIVE "SCHENECTADY"

Diameter of Cylinders 17.038 Diameter of Drivers, inches 62.868 Reversing Lever in 1st notch from

Stroke of Pistons 23.914 Diameter of Exhaust Nozzles inches 3. Center FORWARD

Time and Log Keeper MILLS Date Nov. 20 1895

(For other data see Fuel Log, Injector Log, Feed-Water Log and Calorimeter Log.)

DIRECTIONS:—In case a stop is made during a test, note the time when the drivers cease to turn and enter the same in column 2 with the word "stop" opposite in column 3. When the start is made enter in column 2 on a line below the previous entry the time the drivers begin to turn with the word "start" opposite in column 3. Ring the next regular gong at such a time as will make the running time between it and the previous gong equal to the time of the regular interval between gongs.

Number	Time	Diff. of Time	Driver Counter	Diff. of Driver Counter	Supporting Wheel Counter	Diff. of Supporting Wheel Counter	Dynamometer	Pressure in R. F. Brake	Pressure in L. F. Brake	Pressure in R. R. Brake	Pressure in L. R. Brake	Boiler Pressure	Dry-Pipe Pressure	Water in Gauge-Glass	Temperature in Smoke-Box	Draft, inches of Water	Temperature of Laboratory	Barometer	Galvanometer		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
1	1:45		2692394					0	0	0	0	128	125	4.75		2.6	71	292			
2	1:50	5										134	126								
3	1:55	5	2694806	2412				0	0	0	0	134	126			2.8	74	292			
4	2:00	5										127	125			2.6					
5	2.05	5	2697302	2496				0	0	0	0	127	123			2.4	74	292			
6	2:10	5										129	125			2.4					
7	2:15	5	2699826	2524				0	0	0	0	128	125		659	2.6	76	292	1.8		
8	2:20	5										128	123			2.6					
9	2:25	5	2702316	2490				0	0	0	0	128	125			2.4	76	292			
10	2:30	5										128	125			2.8					
11	2:35	5	2704795	2479				0	0	0	0	130	126			2.8	76	292			
12	2:40	5										127	125			2.6					
13	2:45	5	2707292	2497				0	0	0	0	130	125		650.	2.6	76	292	1.82		
14	2:50	5										126	123			2.0					
15	2:55	5	2709789	2497				0	0	0	0	128	125			2.6	76	292			
16	3:00	5										129	125			2.6					
17	3:05	5	2712285	2496				0	0	0	0	130	125			2.6	77	292			
18	3:10	5										128	125			2.6					
19	3:15	5	2714766	2481				0	0	0	0	129	125		612.	2.6	77	292	1.76		
20	3:20	5										129	125			2.4					
21	3:25	5	2717274	2508				0	0	0	0	129	125			2.6	76	292			
22	3:30	5										127	125			2.6					
23	3:35	5	2719757	2483				0	0	0	0	127	123			2.6	76	292			
24	3:40	5										130	125			2.8					
25	3:45	5	2722255	2498				0	0	0	0	128	125		650.	2.8	77	292	1.8		
26	3:50	5										128	128								
27	3:55	5	2724748	2493				0	0	0	0	130	127			2.8	76	292			
28	4:00	5										129	123			2.8					
29	4:05	5	2727266	2518				0	0	0	0	128	124			2.8	76	292			
30	4:10	5										129	124			2.9					
31	4:15	5	2729753	2487				0	0	0	0	132	126	4.75	650		76	292	1.73		
Totals		150		37359								3994	3872		3221	73.3	1210	4672	8.91		
Averages				2490.6								128.8	124.9		644	2.62	75.6	29.2	1.78		

FROM PARTIAL LOGS.

I.	Pounds of feed-water delivered to injectors	21687.6	XIII.	Quality of steam in Dome	98.43%
II.	Temperature of feed-water	56.4°F	XIV.		
III.	Times injector was started	2.	XV.	Kind of fuel	Brazil Block Coal.
IV.	Minutes injector was in action	146.	XVI.	Pounds of coal fired	3339.
V.	Times safety valves opened	0	XVII.	Per cent of water in coal as fired	2.
VI.	Minutes " " were open	0	XVIII.	Pounds of dry coal	3272.22
VII.	Pounds of water lost by injector overflow	15.	XIX.	Pounds of ash	231.5
VIII.	Pounds of water delivered to boiler and presumably evaporated	21672.6	XX.	" " combustible	3040.72
IX.	Pounds lost by safety valve	0.	XXI.	" " cinders caught in smoke-box	135.75
X.	" " " calorimeter	51.75	XXII.		
XI.	Other loss from boiler		XXIII.		
XII.	Pounds to be deducted from weight delivered to boiler to give weight of steam used by engine.	51.75	XXIV.		

NOTES: No water in overflow at Start. 15 Pounds at Finish.
Engine Position 4.

FIG. 55.

As the readings were taken they were entered on the appropriate logs, facsimiles of which are shown in Figs. 55–60. The card log and the summary sheet (Figs. 59 and 60) were filled out later as the test was worked up, the card log containing all the indicator-card measurements, while the summary sheet, as its name implies, comprises all

FEED-WATER LOG.

Test No...45-J....Series..A.......... Locomotive..."Schenectady"........

Observer....Harlan.................. Date...Nov..20... ...1895.....

LEFT BARREL.				RIGHT BARREL.					
Check.	Time.	Check.	Time.	Check.	Time.	Temp.	Check.	Time.	Temp.
✓	1:47	✓	3:16	✓	1:50	58	✓	3:19	
✓	1:55	✓	3 21	✓	2:00		✓	3:23	
✓	2:04	✓	3:25	✓	2.08		✓	3:27	
✓	2:10	✓	3:30	✓	2:12		✓	3:32	56.
✓	2:14	✓	3:35	✓	2:16	57.	✓	3.37	
✓	2:18	✓	3.40	✓	2:20		✓	3.43	
✓	2:22	✓	3:46	✓	2:24		✓	3:50	
✓	2:27	✓	3:53	✓	2 30	57	✓	3.55	
✓	2:32	✓	3:57	✓	2:34		✓	3:59	56.
✓	2:36	✓	4.02	✓	2:38		✓	4:05	
✓	2:41	✓	4:09	✓	2 44	56	✓	4:11	
✓	2:47	✓	4:13	✓	2:50		✓	4:15	56.
✓	2:52			✓	2 55				
✓	2.59			✓	3.01	56.			
✓	3.04			✓	3 06				
✓	3:09			✓	3.13	56.			

Left Barrel ---Times Emptied..28 Calibrated Wt..389..Wt. of Water.10892..

Right Barrel ---Times Emptied..27.4..Calibrated Wt..394..Wt. of Water.10795.6

Average Temperature...56.4°F Total Wt. of Water.21687.6

Note...Only .4..of last Barrel used..

FIG. 56.

the final calculated and observed results from the test, together with the constants and dimensions of the engine.

28. Data.—The tables at the end of this chapter present the results of forty-four efficiency tests made upon locomotive Schenectady No. 1. These tests are not all that were run during the period

covered, but they comprise all the later tests and have been chosen as those concerning the reliability of which no doubt can be entertained. Only final results are given, which, in the case of observed data, are generally the average of observations made at five-minute intervals and checked either by two different observers or by one observer and some form of automatic recording-instrument. In

ENGINEERING LABORATORY. PURDUE UNIVERSITY.

FUEL LOG

Of Test No. 45-1-A of Locomotive Boiler in connection with "Schenectady" Test No. 45-1-A

Observers: Spiker. (a) Kind of Fuel Brazil Block Coal Date November 20, 1895

1	2	3	4	5	1	2	3	4	5
Time of Gong.	Weight of Full Box.	Weight of Empty Box.	Weight of Coal.		Time of Gong.	Weight of Full Box.	Weight of Empty Box.	Weight of Coal.	
1:45					2.55				
	332	94	238.			323.	93.	230.	
	315.	93	222.		3:00				
	327.	93.	234.			330	93	237.	
1.50					3:05				
1.55					3:10				
2:00					3:15				
2.05						348.	93.	255.	
	323.	93.	230.		3:20				
2:10					3:25				
	331.	93.	238.			338.	93.	245.	
2.15					3:30				
2.20					3:35				
	335.	93.	242.			346.	93.	253.	
2:25					3:40				
2:30					3:45				
	329.	93.	236.		3.50				
2:35						322.	93.	229.	
2:40					3:55				
	343.	93.	250.		4:00				
2:45					4:05				
2:50					4:10				
					4:15				

a. Pounds weighed out 3339.
b. " " back 0.
c. Weight of coal fired 3339.
d. Per cent. of water in coal fired 2.

e. Weight of dry coal fired 3272.22
f. " " ash 231.5
g. " combustible 3040.72
h. " dry cinders caught in smoke box 135.75

FIG. 57.

the case of derived results the calculations have been made by two or more independent workers and carefully checked.

That the facts may be more readily apprehended the results are divided into nineteen tables grouped under four heads, namely, General Conditions (I–III), Boiler Performance (IV–VII), Engine Performance (VIII–XVI), Locomotive Performance (XVII–XIX).

Each test is designated by Consecutive Number, by Laboratory Symbol, and by Date. The Laboratory Symbol tells at a glance the speed in miles per hour, the position of the reverse-lever, and the series to which the test belongs. Thus 15–1–A denotes a test run at a speed of fifteen miles per hour, with the reverse-lever in the first notch forward of the center and under the conditions of Series A. The conditions of valve-setting and proportions which

ENGINEERING LABORATORY PURDUE UNIVERSITY.

Test for Determining the Quality of Steam

BY USE OF THROTTLING CALORIMETER No.

Tests made in connection with LOCOMOTIVE TEST *No.* 45-1-A

Made by KINNBIRD *Date* NOVEMBER 20, 1895.

Gong Number	Time	Difference	Steam Pressure by Gauge	Observed Temperature of Steam	Pressure in Calorimeter by Gauge	Observed Temperature in Calorimeter	NOTES
1	1:45		128.		5.	280.	
3	1:55	10	134.		5.	280.	
5	2:05	10	127.		5.	279.	
7	2:15	10	128.		5.	279.5	
9	2:25	10	128.		5.	279.	
11	2:35	10	130.		5.	279.5	
13	2:45	10	130.		5.	279.5	
15	2:55	10	128.		5.	279.5	
17	3:05	10	130.		5.	279.5	
19	3:15	10	129.		5.	279.5	
21	3:25	10	129.		5.	279.5	
23	3:35	10	127.		5.	279.5	
25	3:45	10	128.		5.	279.5	
27	3:55	10	130.		5.	280.	
29	4:05	10	128.		5.	279.5	
31	4:15	10	132.		5.	279.5	
		150	2066		80.	4472.5	
			129.1		5.	279.5	

Barometric Pressure 14.308

Absolute Steam Pressure 143.408

Temperature corresponding with Steam Pressure 354.72

Absolute Pressure in Calorimeter 19.308

Temperature corresponding with Pressure in Calorimeter 226.07

(a) QUALITY OF STEAM IN

Per cent of Priming 1.57

Degrees Superheated

FIG. 58.

distinguished the various series are listed under Constants. Throughout this set of tables the tests are arranged in series, with low speeds and early points of cut-off first in each series, and for convenience in comparison and reference, the Consecutive Numbers and Laboratory Symbols are reprinted with each table. In succeeding chapters it has been found convenient in some instances to

select certain data and present them rearranged to suit the matter in hand. In all such cases the Laboratory Symbol affords a ready means of identification.

The following concerns the several items appearing in the tables and the methods of computation:

ENGINEERING LABORATORY. PURDUE UNIVERSITY.

PRESSURES AS OBTAINED FROM INDICATOR CARDS.

Test No. 45-1. Series A of Locomotive Schenectady Made November 20, 1895

Cards Measured by DeWolfe and Talbot

Recorded by DeWolfe Date December 20 1895

Spring of Cards 80 Pounds.

	Right Side											
	Head-End						Crank-End					
No. of Card.	Initial.	Cut-off.	Release.	Compression.	Least Back.	M.E.P.	Initial.	Cut-off.	Release.	Compression.	Least Back.	M.E.P.
2	122	64	16	23	3.3	22.7		No Card.				
4		No Card.					112	66	17	22	3	22.8
6	103	61	15	24	2.0	20.1	110	67	17	20	2.	21.6
8	110	62	16	26	30	20.8	104	67	17	22	3	22.6
10	112	62	16	23	2.5	20.	113	73	16	21	3.	22.8
12	113	65	16	23	20	21.4	113.	68	18	24	3.	22.3
14	110	62	14	20	2.0	20.7	112	72	16	20	2.8	22.5
16	110	64	14	14	1.7	20.7	104	70	18	22	3.	21.4
18	110	65	14	21	2	20.7	114	75	18	26	2.5	22.1
20	110	61	14	27	2	21.4	104	71	18	25	3.3	22.4
22	112	66.5	15	23	2.	21.8	116	74	18	25	3	23.1
24	104	67	14	16	2	21.3	106	74	17	24	4.	22.7
26	115	67.	14	25	2.5	20.6	113	64	18	25	3.	22.7
28	111	65.	14	22	2.	19.6	108	71	17	22	3.	22.4
30	113	64.	16.	25	3.	20.8	111	73	18	25	2.5	24.2
Totals	1560.	905.5	208.	317	32.	293.1	1555.	990.	243.	322	41.1	316.1
Average	111.4	64.7	14.9	22.6	2.2	20.9	111.1	70.7	17.3	23.	2.9	22.6

	Left Side											
	Head-End						Crank-End					
No. of Card.	Initial.	Cut-off.	Release.	Compression.	Least Back.	M.E.P.	Initial.	Cut-off.	Release.	Compression.	Least Back.	M.E.P.
2	125	66	15	25	2.5	24.8	126	75	20	26	4	27
4	116	65	14	23	3	22.3	115	74	17	25	2.5	24.4
6	113	61	15	24	2.5	21.6	114	70	17	23	2	23.2
8	114	62	15	25	3	22.6	115	70	16	14	2.	24.5
10	116	62	14	20	2.3	22.1	118	68	14	22	3	24.5
12	120	71	16	26	2.5	21.4	124	72	18	25	3	23.6
14	115	66	14	23	2	22.8	118	76	17	24	2.5	23.3
16	123	64	14	27	3.	22.6	118	73	17	23	3.	23.3
18	118	62	13	26	2.5	22.2	120	72	18	22	2.5	22.6
20	114	61	14	20	2.5	21.2		No Card				
22	121	66	15	26	3	22.1	116	71	17	23	2.5	23.4
24		No Card.					112	70	17	24	3.	22.8
26	118	65	15	24	3.5	22.3	116	71	14	22	3	23.7
28	114	66	15	24	3	21.	114	64	16	23	2.5	22.2
30	121	68	16	25	3.	23.2	114	74	17	22	3.5	24.3
Totals	1653.	905.	205.	348.	38.3	312.7	1645.	1005.	245.	323	34.	333.8
Average	118.	64.6	14.6	24.9	2.7	22.3	117.5	71.6	17.5	23.1	2.4	23.8

Barometric Pressure, lbs. 14.308 Absolute Pressure at Release 30.383

Initial Absolute Pressure 128.808 Absolute Pressure at Compression 37.708

Absolute Pressure at Cut-off 82.208 Absolute Mean Back-Pressure 16.968

Fig. 59.

Item 1.—Consecutive Number.
Item 2.—Laboratory Symbol.
Item 3.—Date of Running Test.
Item 4.—Duration of Test in Minutes.

FORM 1894.

ENGINEERING LABORATORY. **SUMMARY OF RESULTS.** PURDUE UNIVERSITY.

Test No. 95-1-A made Nov. 20, 1895 on Locomotive "Schenectady"

Calculated by Talbot and De Wolfe Checked by R. S. Miller

CONCERNING THE LOCOMOTIVE.

Total weight, lbs.	85,000.	Circumference of drivers, ft.	16.359	Total area of heating surface, sq. feet	1219.9
Weight on 4 drivers	56,000.	Dia. of boiler, inches	52.	Area of grate surface sq. feet	17.5
Dia. of cylinders	17.038	Dia. of tubes, inches	2.	Diameter of exhaust nozzle, inches	3.
Stroke of piston	23.914	Length of tubes, feet	11.5		
Dia. of drivers	62.968	No. of tubes	200.		

CONSTANTS FOR TEST.

Reversing lever in 1st notch from center Forward. Tractive—H. P. constant .0004956

Tractive force assumed to be necessary to draw one ton on level track at speed of test.

	Right Side H. E.	Right Side C. E.	Left Side H. E.	Left Side C. E.	Average.
Piston displacement, cu. ft.	3.1616556	3.0637365	3.1996822	3.0519549	3.1067573
Clearance, per cent. of piston displacement	9.103	10.096	10.086	9.88	9.779
Indicated H. P. constants	.0137963	.0133688	.0137491	.0133176	.0135567
Admission, per cent. of stroke	1.79	2.69	1.79	2.39	2.152
Cut-off " "	21.36	23.09	22.91	22.36	22.432
Release " "	73.02	73.09	75.07	72.18	73.39
Beginning of compression, per cent of stroke	21.89	29.11	20.82	23.96	22.695
Volume at cut-off, cu. ft.	.9631351	1.0136679	1.0911589	.9839509	1.0007276
Volume at release " "	2.5980273	2.5955351	2.6821939	2.5099392	2.582535
Volume at beginning of compression, cu. ft.	.9798919	1.0969998	.9739902	1.0327815	1.0081909

RESULTS FROM THE INDICATOR.

	Right Side H. E.	Right Side C. E.	Left Side H. E.	Left Side C. E.	Average.
A Initial pressure	111.9	111.1	118.0	117.5	114.5
B Pressure at cut-off	69.7	70.7	69.6	71.6	67.9
C Pressure at release	19.9	17.3	19.6	17.5	16.023
D Back pressure	2.2	2.99	2.7	2.8	2.66
E Pressure at the beginning of compression	22.6	23.	24.9	23.1	23.4
F Mean effective pressure	20.9	22.6	22.3	23.8	22.4

RESULTS FROM LOGS OF TEST.

A₁ Duration of test in minutes	150.	I₁₇ Draft, inches of water	2.68	Q Quality of steam in dry-pipe	
B₂ Total revolutions	37359.	J₁₆ Temperature of smoke box	699.35	R₁ Kind of fuel	Brazil Block Coal.
C Revolutions per minute	249.06	K₁₅ Temperature of laboratory	75.6	S₂ Pounds of dry coal	3272.22
D₂ Dynamometer reading		L₇ Temperature of feed water	56.4	T₁ Pounds of dry ash	231.5
E₁₃ Boiler pressure	128.89	M VIII Pounds of water delivered to boiler and presumably evaporated	21,672.6	U₂ Pounds of combustible	3040.72
F₁₄ Dry-pipe pressure	129.9	N XII Pounds water lost from boiler	51.75	V₁ Pounds of cinders in smoke smoke box	135.75
G₁₀ Barometric pressure	14.308	O Pounds steam supplied engine	21,620.85		
H₁₂ Weight of water in gauge-glass		P Quality of steam in boiler	98.43		

CALCULATED RESULTS.

	Right Side H. E.	Right Side C. E.	Left Side H. E.	Left Side C. E.	Total.
1 Indicated horse power	71.8196	75.2491	76.3352	78.9409	302.3398
2 Weight of steam at cut-off	.1759838	.1977666	.1899805	.1939169	.7576478
3 Weight of steam at release	.1869399	.1967398	.1906178	.1942142	.7685005
4 Weight of steam during compression	.0876191	.09953939	.09811085	.0935163	.36777569

5 Re-evaporation per rev.	.0118533	16 Distance equivalent to total revolution in miles	115.714	27 Equivalent ton-miles	
6 Re-evaporation per H. P. per hour	.585766	17 Miles per hour	46.286	28 Pounds of coal per indicated H. P. per hour	9.3291
7 Weight of steam per rev. by tank	.578732	18 Dynamometer H. P.		29 Pounds of coal per mile run	28.2785
8 Weight of mixture in cylinder, per rev.	.996507	19 Friction of engine in H. P.		30 Pounds of coal per ton-mile	
9 Per cent. of mixture accounted for as steam at cut-off	79.941	20 Friction of engine, per cent. of I. H. P.		31 Pounds of water evaporated per pound of coal	6.6232
10 Per cent. of mixture accounted for as steam at release	81.183	21 Dynamometer work in foot tons, per pound of steam by tank		32 Equivalent evaporated from and at 212° F.	7.9001
11 Pounds of steam per indicated H. P. per hour by indicator	19.47569	22 Dynamometer work in foot tons per pound of dry coal		33 B. T. U. taken by boiler per hour	9,987,089.
12 Pounds of steam per indicated H. P. per hour by tank	28.60378	23 Equivalent weight of train in tons		34 B. T. U. taken up per pound of combustible	8211.
13		24 Equivalent number of loaded freight cars of 33 tons each		35 B. T. U. taken up by boiler per H. P. per hour	33,032.69
14		25		36 Pounds coal per foot of grate per hour	74.7931
15		26		37 Indicated H. P. per foot of grate	17.276
				38	

FIG. 60.

Item 5.—Valve-setting and Proportions. The Roman numerals here refer to the table of valve proportions under Constants (Paragraph 25).

Item 6.—Approximate Cut-off is a predetermined condition for the test.

Item 7.—Reverse-Lever, notch Forward of Center.

Item 8.—Throttle Position.

Item 9.—Draw-bar Pull is derived from the known value of the work done in cylinders and of that lost by friction = Item 134 − Item 136.

Item 10.—Total Revolutions is the record from a counter attached to the rear driver, which was read every five minutes as a check on the regularity of running. In addition there was a Boyer railway speed-indicator reading in miles per hour.

Item 11.—Revolutions per Minute = Item 10 ÷ Item 4.

Item 12.—Miles Run Equivalent to Total Revolutions = Item 10 × Circumference of Driver in feet ÷ 5280.

Item 13.—Miles per Hour = Item 12 ÷ (Item 4 ÷ 60).

Item 14.—Feed-water Temperature by Thermometer.

Item 15.—Laboratory Temperature by Thermometer.

Item 16.—Smoke-box Temperature is the record obtained from a copper-ball pyrometer used as follows: A copper ball of known weight was held in the smoke-box till it had presumably attained the temperature of its surroundings, usually about forty minutes, and was then allowed to roll quickly down a tube into a water calorimeter. The original temperature of the ball was computed from data thus obtained.

Item 17.—Boiler Pressure by Gauge is the record of readings from an ordinary dial-gauge at intervals of five minutes as checked by a recording-gauge.

Item 18.—Atmospheric Pressure in Pounds.

Item 19.—Absolute Boiler Pressure = Item 17 + Item 18.

Item 20.—Dry-pipe Pressure by Gauge.

Item 21.—Times Injectors were Started includes the record for both right-hand and left-hand injectors.

Item 22.—Minutes one or both injectors were in action. Shows the regularity of feed-water supply.

Item 23.—Draft is assumed to be the difference in pressure between

the atmosphere and the interior of the smoke-box, as measured by a U tube in inches of water. This tube was fastened securely to the laboratory wall, having one end in pipe connection with the interior of the smoke-box and the other open to the atmosphere. The pipe entered the smoke-box at a point marked *C*, Fig. 35, Chapter III. Subsequent experiment has shown that this is not the point of least pressure, and that to obtain the actual greatest draft these values should be multiplied by a factor of 1.3. (See Chapter XI.)

Item 24.—Water Delivered to Boiler and Presumably Evaporated is the actual weight of water delivered to the injectors minus the weight caught from the injector overflow. The water was measured by means of two carefully calibrated barrels, one of which was filled while the other was being emptied into the receiving-tank. A water-meter in the supply-line served as a check to prevent any gross error in measuring.

Item 25.—Water Lost from Boiler is the weight of steam used by the calorimeter as determined by previous experiment. In no case were boiler leaks permitted.

Item 26.—Steam Supplied Engine = Item 24 − Item 25.

Item 27.—Water Evaporated by Boiler per Hour = Item 24 ÷ (Item 4 ÷ 60).

Item 28.—Steam Used by Engine per Hour = Item 26 ÷ (Item 4 ÷ 60).

Item 29.—Quality of Steam as indicated by the throttling calorimeter.

Item 30.—Water Evaporated per Square Foot Heating Surface = Item 24 ÷ 1214.4.

Item 31.—Water Evaporated per Square Foot Grate Surface = Item 24 ÷ 17.25.

Item 32.—Dry Coal Fired is the total weight of coal less the amount of accidental moisture, as found by sample taken during each test, a correction which seldom exceeded one per cent.

Item 33.—Dry Ash Caught in the Ash-pan during the Test. Inasmuch as under locomotive conditions a large per cent of the non-combustible material goes out of the stack, the weight of ash was not used to determine the amount of combustible fired.

Item 34.—Dry Cinders Caught in the Smoke-box.

Item 35.—Dry Coal fired per hour = Item 32 ÷ (Item 4 ÷ 60).

Item 36.—Combustible Fired per Hour = Item 35 × 0.9. Throughout these tests the fuel used was Brazil block coal, mined at Brazil, Indiana. Frequent analysis of the chemical composition showed a very uniform content of combustible, the average being 90 per cent.

Item 37.—Dry Coal per Square Foot Heating Surface per Hour = Item 35 ÷ 1214.4.

Item 38.—Dry Coal per Square Foot Grate Surface per Hour = Item 35 ÷ 17.25.

Item 39.—Same as Item 29.

Item 40.—B. T. U. taken up by each pound of Steam is the result obtained by the formula

$$\text{B. T. U.} = q - q_0 + xr,$$

where q = the heat in one pound of water at boiler temperature, q_0 = heat in one pound of water at temperature of the feed-water, x = quality of steam as given in column 39, and r = the latent heat of evaporation at boiler pressure. (Peabody's Steam-tables used for all computations involving thermal quantities.)

Item 41.—B. T. U. taken up by Boiler per Minute = Item 40 × Item 27 ÷ 60.

Item 42.—B. T. U. per Pound of Coal = Item 40 × Item 27 ÷ Item 35.

Item 43.—B. T. U. per Pound of Combustible = Item 40 × Item 27 ÷ Item 36.

Item 44.—Water Evaporated per Pound of Coal = Item 27 ÷ Item 35.

Item 45.—Water Evaporated per Pound of Combustible = Item 27 ÷ Item 36.

Item 46.—Same as Item 27.

Item 47.—Equivalent Evaporation per Hour = Item 27 × Item 4 ÷ 0 latent heat of evaporation at atmospheric pressure (= 965.8 B. T. U.).

Item 48.—Equivalent Evaporation per Square Foot Heating Surface per Hour = Item 47 ÷ 1214.4.

Item 49.—Equivalent Evaporation per Square Foot Grate Surface per Hour = Item 47 ÷ 17 25

Item 50.—Equivalent Evaporation per Pound of Dry Coal = Item 47 ÷ Item 35.

Item 51.—Equivalent Evaporation per Pound of Combustible = Item 47 ÷ Item 36.

Item 52.—Boiler Horse-Power = Item 47 ÷ 34.5.

Items 53 to 102.—Events of Stroke and Pressures above Atmosphere from Indicator-cards need no explanation. The same point on the card was used to determine the event of stroke and the corresponding pressure.

Items 103 to 117.—Weight of Steam present at the Several Events of the Stroke is the result obtained by the formula

$$w = \frac{v}{s},$$

where w = the weight in pounds, v = the per cent of stroke plus clearance per cent multiplied into the cylinder volume, and s = the volume of one pound of steam at the pressure above atmosphere as measured on the card.

Items 118 to 122.—Indicated Horse-power from the Card Measurements.

Item 123.—Steam per I. H. P. per Hour = Item 28 ÷ Item 122.

Item 124.—Steam per I. H. P. per Hour by Indicator = (Item 112 − Item 117) × Item 11 × 60 ÷ Item 122.

Item 125.—Dry Coal per I. H. P. per Hour = Item 35 ÷ Item 122.

Item 126.—Weight of Steam per Revolution by Tank = Item 26 ÷ Item 10.

Item 127.—Weight of Mixture in Cylinder per Revolution = Item 126 + Item 117.

Item 128.—Per Cent of Mixture present as Steam at Cut-off = Item 107 ÷ Item 127.

Item 129.—Per Cent of Mixture present as Steam at Release = Item 112 ÷ Item 127.

Item 130.—Reëvaporation per Revolution = Item 112 − Item 107, when Item 112 > Item 107.

Item 131.—Condensation per Revolution = Item 107 − Item 112, when Item 107 > Item 112.

Item 132.—Reëvaporation per I. H. P. per Hour = Item 130 × Item 11 × 60 ÷ Item 122.

Item 133.—Condensation per I. H. P. per Hour = Item 131 × Item 11 × 60 ÷ Item 122.

Item 134.—Draw-bar Pull Equivalent to Average M. E. P. The

D. H. P. = draw-bar pull $\times$ .0004956 $\times$ R. P. M.

and

I. H. P. = average M. E. P. $\times$ .0542 $\times$ R. P. M.

Assumed that all the I. H. P. was transmitted to the draw-bar without friction losses. Then, these expressions are equal and their second terms may be equated. Performing the operation indicated and solving will give Draw-bar pull = M. E. P. $\times$ 109.419. Item 134 therefore = Item 102 $\times$ 109.419. It was found impracticable to get the correct observed value of draw-bar stress, and therefore it was assumed that at equal cut-offs, friction was constant at all speeds. (See Chapter XIX.) Items 135 to 139 are computed on this basis.

Item 135.—Friction Expressed as M. E. P. is assumed constant for all tests at same cut-off.

Item 136.—Friction Expressed as Draw-bar Pull = Item 135 $\times$ 109.419.

Item 137.—Assumed Friction Horse-power = Item 135 $\times$ .0542 $\times$ Item 11.

Item 138.—Draw-bar Horse-power = Item 122 − Item 137.

Item 139.—Mechanical Efficiency = Item 138 ÷ Item 122.

Item 140.—I. H. P. per Square Foot Heating Surface = Item 122 ÷ 1214.4.

Item 141.—I. H. P. per Square Foot of Grate Surface = Item 122 ÷ 17.25.

Item 142.—Steam Used per Dynamometer Horse-power per Hour = Item 28 ÷ Item 138.

Item 143.—Coal Used per D. H. P. per Hour = Item 35 ÷ Item 138.

Item 144.—B. T. U. Used by Engine per Minute = Item 28 $\times$ Item 40 ÷ 60.

Item 145.—B. T. U. Used by Engine per I. H. P. per Minute = Item 144 ÷ Item 122.

Item 146.—B. T. U. Used by Engine per D. H. P. per Minute = Item 144 ÷ Item 138.

Item 147.—Coal Used per Mile Run = Item 32 ÷ Item 12.

Item 148.—D. H. P. per Square Foot Heating Surface = Item 138 ÷ 1214.4.

Item 149.—D. H. P. per Square Foot Grate Surface=Item 138 ÷ 17.25.

Item 150.—Water Evaporated from and at 212° F. per Hour= Item 47.

Item 151.—Coal fired per Hour=

$$\frac{\text{Item } 150}{10.08 - .000244 \times \text{Item } 150}.$$

Item 152.—Coal burned per Square Foot of Grate per Hour=Item 151 ÷ 17.25.

Item 153.—Water per Pound of Coal=Item 27 ÷ Item 151.

Item 154.—Equivalent Evaporation per Pound of Coal=Item 150 ÷ Item 151.

Item 155.—Coal per I. H. P. per Hour=Item 151 ÷ Item 122.

Item 156.—Coal per D. H. P. per Hour=Item 151 ÷ Item 138.

Item 157.—B. T. U. Taken up by Boiler per Pound of Coal= Item 40 × Item 153.

Table XIX., of Locomotive Performance based on uniform firing conditions, is composed of several items so corrected as to eliminate the effect of irregularities in firing. In this one table the total weight of coal fired per hour, which is a factor in all these items, is not that which was found by experiment, but was calculated from the known weight of water evaporated per hour by use of the formula

$$C = \frac{W}{10.08 - .000244W},$$

where C is the pounds of coal fired per hour, and W is the total pounds of water evaporated from and at 212° per hour. The significance of this formula has been developed in the following chapter, pages 150 and 151. For the purposes of this table, W, Item 153, was taken from the experimental results, and C, Item 151, computed. Items 151 to 155 are calculated, using the value C instead of the experimental results.

The coal used in these tests was the same throughout, namely, Indiana block from the Brazil field, a light fuel which burns freely to a fine ash. The following is the average result of four analyses:

Per cent fixed carbon	51.06
Per cent volatile matter	39.26
Per cent combined moisture	3.14
Per cent ash	6.54
Total	100.00
B. T. U. per pound of dry coal	13,000

It has been shown by tests made subsequent to those herein discussed that 0.8 of a pound of first-class Pittsburg or West Virginia coal will give in locomotive service approximately the same results as one pound of Brazil block.

The record of the observed and calculated results is as follows:

TABLE I.

GENERAL CONDITIONS.

Number.	Laboratory Symbol.	Date.	Duration of Test, Minutes.	Valve Setting and Proportions (see Constants).	Approximate Cut-off, Per Cent of Stroke.	Reverse-lever Notch Forward of Center.	Throttle Position.	Drawbar Pull.
1	2	3	4	5	6	7	8	9
1	15–1$_b$–V	Nov. 23, '94	190	I	25	1st	Fully open	4227
2	25–1–V	Nov. 26, '94	240	I	25	1st	Fully open	3657
3	35–1–V	Dec. 4, '94	140	I	25	1st	Fully open	2796
4	55–1–V	Dec. 18, '95	120	I	25	1st	Fully open	1544
5	15–1–A	Dec. 12, '94	240	II	25	1st	Fully open	4180
6	25–1–A	Dec. 14, '94	255	II	25	1st	Fully open	2893
7	35–1–A	Dec. 17, '94	180	II	25	1st	Fully open	2600
8	45–1–A	Nov. 20, '95	150	II	25	1st	Fully open	1870
9	55–1–A	Nov. 25, '95	120	II	25	1st	Fully open	1342
10	15–2–A	Nov. 13, '95	180	II	35	2d	Fully open	6409
11	25–2–A	Oct. 25, '95	180	II	35	2d	Fully open	5259
12	35–2–A	Dec. 19, '94	180	II	35	2d	Fully open	4157
13	45–2–A	Nov. 18, '95	140	II	35	2d	Fully open	3132
14	55–2–A	Nov. 22, '95	68	II	35	2d	Fully open	2431
15	25–3–A	Nov. 1, '95	122.5	II	45	3d	Fully open	6666
16	35–3–A	Nov. 15, '95	120	II	45	3d	Fully open	5050
17	15–9–A	Nov. 6, '96	150	II	80	9th	Partly open	7224
18	35–2–B	Jan. 14, '95	170	III	35	2d	Fully open	2756
19	35–2–C	Jan. 23, '95	120	IV	35	2d	Fully open	5100
20	35–2–E	Jan. 16, '95	180	V	35	2d	Partly open	2722
21	35–2–F	Jan. 21, '95	180	VI	35	2d	Partly open	2851
22	15–1–G	Nov. 9, '96	180	VII	25	1st	Fully open	4704
23	35–1–G	Nov. 20, '96	170	VII	25	1st	Fully open	2860
24	35–1$_b$–G	Dec. 2, '96	170	VII	25	1st	Fully open	2903
25	55–1–G	Nov. 23, '96	120	VII	25	1st	Fully open	1796
26	35–2–G	Nov. 13, '96	160	VII	35	2d	Fully open	4467
27	35–3–G	Dec. 4, '96	140	VII	45	3d	Partly open	3423
28	15–9–G	Nov. 12, '96	160	VII	80	9th	Partly open	7473
29	15–1–H	Dec. 9, '96	180	VIII	25	1st	Fully open	4418
30	35–1–H	Dec. 18, '96	160	VIII	25	1st	Fully open	2857
31	55–1–H	Feb. 11, '97	120	VIII	25	1st	Fully open	1783
32	35–2–H	Dec. 16, '96	120	VIII	35	2d	Fully open	3833
33	35–2$_b$–H	Feb. 10, '97	160	VIII	35	2d	Partly open	2857
34	35–3–H	Dec. 14, '96	120	VIII	45	3d	Partly open	3316
35	15–9–H	Dec. 11, '96	180	VIII	80	9th	Partly open	7114
36	15–1–I	March 27, '97	270	IX	25	1st	Fully open	
37	35–1–I	March 29, '97	170	IX	25	1st	Fully open	
38	55–1–I	April 1, '97	120	IX	25	1st	Fully open	
39	15–9–I	March 26, '97	190	IX	80	9th	Partly open	
40	15–4–J	April 14, '97	300	X	25	4th	Fully open	
41	35–4–J	April 13, '97	180	X	25	4th	Fully open	
42	15–15–J	April 9, '97	160	X	80	15th	Partly open	
43	15–2–K	April 27, '97	300	XI	35	2d	Fully open	
44	35–2–K	April 24, '97	180	XI	35	2d	Fully open	

TABLE II.

GENERAL CONDITIONS—*Continued.*

Number.	Laboratory Symbol.	Speed.				Temperature, Degrees F.		
		Total Revolutions.	Revolutions per Minute.	Miles Run Equivalent to Total Revolutions.	Miles per Hour.	Feed-water.	Laboratory.	Smoke-box.
1	2	10	11	12	13	14	15	16
1	15-1_b-V	14,891	78.37	46.12	14.56	53.76	73.93	549.50
2	25-1-V	29,174	121.56	90.36	22.59	53.85	69.62	605.60
3	35-1-V	26,111	186.51	80.88	34.66	51.90	69.00	633.00
4	55-1-V	36,080	300.67	111.75	55.88	58.10	75.70	666.60
5	15-1-A	19,367	80.69	59.99	15.00	53.20	65.46	553.20
6	25-1-A	33,024	129.50	102.29	24.07	53.75	67.34	567.0
7	35-1-A	34,373	190.96	106.47	35.49	53.17	74.91	628.10
8	45-1-A	37,359	249.06	115.71	46.29	56.40	75.60	644.30
9	55-1-A	36,693	305.75	113.65	56.83	56.00	75.90	675.40
10	15-2-A	13,878	77.10	42.98	14.33	55.19	79.30	620.60
11	25-2-A	22,973	127.63	71.15	23.72	56.00	84.80	696.30
12	35-2-A	34,204	190.02	105.94	35.31	52.54	72.20	720.30
13	45-2-A	34,454	246.10	106.72	45.74	55.00	83.70	740.50
14	55-2-A	20,808	306.00	64.45	56.87	58.40	76.40	754.60
15	25-3-A	15,819	129.10	49.00	24.00	53.30	76.40	762.10
16	35-3-A	22,054	183.78	68.31	34.15	55.50	76.70	798.10
17	15-9-A	11,886	79.24	36.82	14.73	55.62	78.50	724.20
18	35-2-B	32,029	188.41	99.20	35.01	50.00	65.90	664.30
19	35-2-C	22,746	189.55	70.45	35.23	51.68	70.80	737.70
20	35-2-E	34,947	194.15	108.24	36.08	51.88	70.60	652.00
21	35-2-F	33,673	187.07	104.30	34.76	52.72	71.50	653.10
22	15-1-G	14,589	81.05	45.18	15.06	53.33	68.70	582.50
23	35-1-G	32,361	190.36	100.23	35.38	55.00	74.40	684.70
24	35-1_b-G	32,341	190.24	100.17	35.35	52.47	77.60	618.30
25	55-1-G	36,300	302.50	112.43	56.22	54.17	79.40	
26	35-2-G	29,952	187.20	92.77	34.80	53.55	77.90	654.50
27	35-3-G	26,612	190.09	82.43	35.33	52.62	69.60	719.00
28	15-9-G	12,619	78.87	39.08	14.66	53.13	77.40	724.00
29	15-1-H	14,660	81.44	45.41	15.14	54.63	71.50	570.00
30	35-1-H	30,397	189.98	94.15	35.31	52.59	67.10	655.00
31	55-1-H	35,308	294.23	109.36	54.68	52.08	70.60	695.00
32	35-2-H	23,183	193.19	71.81	35.90	53.01	71.80	
33	35-2_b-H	30,653	191.58	94.94	35.60	50.56	74.30	689.20
34	35-3-H	23,289	194.07	72.13	36.07	53.00	73.50	
35	15-9-H	14,421	80.12	44.67	14.89	53.22	81.10	695.70
36	15-1-I	21,853	80.93	67.69	15.04	52.80	66.40	
37	35-1-I	32,444	190.84	100.49	35.47	56.30	74.50	
38	55-1-I	34,868	290.57	108.00	54.00	54.50	79.50	
39	15-9-I	15,067	79.30	46.67	14.73	51.90	68.00	
40	15-4-J	24,934	83.11	77.23	15.45	53.77		
41	35-4-J	34,308	190.60	106.26	35.42	54.00		
42	15-15-J	12,987	81.17	40.23	14.62	51.10		
43	15-2-K	24,671	82.24	76.41	15.28	56.00		
44	35-2-K	33,722	180.73	104.45	34.82	56.00		

TABLE III.

GENERAL CONDITIONS—*Continued.*

Number.	Laboratory Symbol.	Pressures, Pounds per Square Inch. Boiler by Gauge.	Atmosphere by Barometer.	Absolute Boiler.	Dry Pipe by Gauge.	Injectors. Times Started.	Minutes One or Both in Action.	Draft, Inches of Water.*
1	2	17	18	19	20	21	22	23
1	15-1$_b$-V	127.27	14.54	141.8	130.00	14	129	2.04
2	25-1-V	127.12	14.34	141.4	130.40	7	213	2.60
3	35-1-V	128.87	14.32	143.2	127.20	1	136	3.43
4	55-1-V	128.40	14.38	142.8	123.04	1	120	3.20
5	15-1-A	125.94	14.46	140.4	124.58	22	130	1.72
6	25-1-A	120.04	14.52	134.6	113.37	10	236	1.93
7	35-1-A	129.72	14.62	144.3	127.23	2	180	3.00
8	45-1-A	128.84	14.31	143.1	124.90	2	146	2.68
9	55-1-A	124.91	14.18	139.1	121.29	1	120	2.58
10	15-2-A	129.48	14.39	143.9	125.10	9	149	2.42
11	25-2-A	129.32	14.35	143.7	124.90	1	180	3.37
12	35-2-A	131.65	14.49	146.1	120.67	1	180	4.42
13	45-2-A	126.67	14.29	141.0	122.10	1	140	4.93
14	55-2-A	124.00	14.43	138.4	118.80	1	68.5	4.58
15	25-3-A	127.19	14.47	141.7	123.20	1	122.5	5.45
16	35-3-A	125.28	14.34	139.6	122.08	1	120	7.49
17	15-9-A	122.47	14.34	136.8	74.17	1	150	4.56
18	35-2-B	98.36	14.39	112.8	95.57	8	158	3.28
19	35-2-C	143.28	14.42	157.7	143.08	3	125	5.13
20	35-2-E	128.05	14.49	142.6	95.46	3	175	2.89
21	35-2-F	155.35	14.68	170.1	93.11	9	149	2.57
22	15-1-G	123.58	14.39	138.0	120.54	12	127	1.87
23	35-1-G	125.00	14.55	139.6	123.91	1	170	3.02
24	35-1$_b$-G	128.23	14.56	142.8	126.44	1	170	2.98
25	55-1-G	126.90	14.46	141.4	123.18	1	120	3.57
26	35-2-G	121.04	14.54	135.5	118.34	1	160	4.65
27	35-3-G	125.93	14.34	140.2	86.62	1	140	4.88
28	15-9-G	124.06	14.49	138.6	76.33	1	154	4.76
29	15-1-H	123.55	14.32	137.9	119.97	7	126	1.93
30	35-1-H	121.16	14.57	135.8	117.09	1	160	3.00
31	55-1-H	127.48	14.39	141.9	121.52	1	120	3.44
32	35-2-H	112.04	14.58	126.6	108.36	3	115	4.33
33	35-2$_b$-H	122.64	14.54	137.1	92.97	1	160	3.18
34	35-3-H	116.48	14.29	130.8	74.52	1	120	4.52
35	15-9-H	122.72	14.37	137.1	73.94	1	180	4.99
36	15-1-I	127.95	14.53	142.5	124.43			
37	35-1-I	128.05	14.45	142.5	124.97			
38	55-1-I	125.43	14.47	139.9	122.90			
39	15-9-I	131.61	14.35	145.9	72.91			
40	15-4-J	127.51	14.50	142.0	126.49			
41	35-4-J	131.95	14.42	146.4	124.05			
42	15-15-J	128.94	14.41	143.3	89.61			
43	15-2-K	125.50	14.21	139.7	123.31			
44	35-2-K	126.02	14.15	140.2	124.57			

* Multiply these values by 1.3 to obtain draft at point of maximum draft (in front of diaphragm).

TABLE IV.
BOILER PERFORMANCE.
WATER AND STEAM.

Number.	Laboratory Symbol.	Water Delivered to Boiler and Presumably Evaporated. Lbs.	Water Lost from Boiler. Lbs.	Steam Supplied to Engine. Lbs.	Water Evaporated by Boiler per Hour. Lbs.	Steam Used by Engine per Hour. Lbs.	Quality of Steam in Dome. Moisture. Per Cent.	Water Evaporated per Hour per Square Foot of Heating Surface. Lbs.	Water Evaporated per Hour per Square Foot of Grate Surface. Lbs.
1	2	24	25	26	27	28	29	30	31
1	15-1_b-V	17,789.6	67.6	17,721.0	5,617.8	5,596.4	.63	4.626	325
2	25-1-V	29,475.7	82.3	29,393.4	7,368.9	7,348.4	.68	6.068	427
3	35-1-V	19,899.2	48.6	19,850.6	8,528.2	8,507.4	.63	7.022	494
4	55-1-V	19,418.4	41.6	19,376.8	9,709.2	9,688.4	.88	7.995	563
5	15-1-A	22,100.2	81.6	22,018.6	5,525.1	5,504.7	.49	4.549	320
6	25-1-A	25,894.6	84.5	25,810.1	6,092.8	6,072.9	.78	5.017	353
7	35-1-A	24,367.5	62.8	24,304.7	8,122.5	8,101.6	.83	6.688	471
8	45-1-A	21,672.6	51.7	21,620.9	8,669.0	8,648.3	.87	7.138	502
9	55-1-A	17,882.6	40.6	17,842.0	8,941.3	8,921.0	1.22	7.363	518
10	15-2-A	21,840.6	62.7	21,777.9	7,280.2	7,259.3	.76	5.995	422
11	25-2-A	28,906.0	62.7	28,843.3	9,635.3	9,614.4	.90	7.934	558
12	35-2-A	34,324.5	64.0	34,260.5	11,441.5	11,420.2	1.06	9.422	663
13	45-2-A	29,100.4	47.7	29,052.7	12,471.6	12,451.2	1.24	10.269	722
14	55-2-A	15,911.4	22.9	15,888.5	14,039.5	14,019.2	1.13	11.561	813
15	25-3-A	26,401.0	41.8	26,359.2	12,931.1	12,910.6	1.11	10.648	749
16	35-3-A	29,874.8	40.6	29,834.2	14,937.4	14,917.1	1.11	12.300	865
17	15-9-A	28,522.0	50.0	28,472.0	11,408.8	11,388.8	.94	9.395	661
18	35-2-B	24,554.7	46.5	24,508.2	8,666.3	8,649.9	.69	7.136	502
19	35-2-C	26,008.5	45.6	25,962.9	13,004.2	12,981.3	.70	10.708	754
20	35-2-E	25,696.7	62.1	25,634.6	8,565.6	8,544.9	.68	7.053	496
21	35-2-F	24,333.3	74.1	24,259.2	8,111.1	8,086.4	1.00	6.679	470
22	15-1-G	18,695.7	60.3	18,635.4	6,231.9	6,211.8	1.06	5.132	361
23	35-1-G	26,562.7	57.5	26,505.2	9,375.1	9,354.8	1.14	7.720	543
24	35-1_b-G	26,743.3	58.9	26,684.4	9,438.8	9,418.0	1.40	7.772	547
25	55-1-G	20,760.9	41.0	20,719.9	10,380.4	10,359.9	1.31	8.548	602
26	35-2-G	32,846.0	52.8	32,793.2	12,317.3	12,297.4	1.44	10.143	714
27	35-3-G	27,060.2	47.6	27,012.6	11,597.2	11,576.8	1.62	9.550	675
28	15-9-G	30,872.4	55.7	30,816.7	11,577.1	11,556.2	1.29	9.533	671
29	15-1-H	18,223.4	60.3	18,163.1	6,074.5	6,054.4	1.62	5.002	351
30	35-1-H	24,723.3	52.8	24,670.8	9,271.2	9,251.4	1.49	7.634	537
31	55-1-H	20,513.6	41.0	20,472.6	10,256.6	10,236.3	1.29	8.446	595
32	35-2-H	24,733.3	36.8	24,696.5	12,366.6	12,348.2	1.34	10.183	717
33	35-2_b-H	25,829.0	50.7	25,778.3	9,685.9	9,666.9	1.20	7.976	561
34	35-3-H	23,385.7	38.2	23,347.5	11,692.9	11,673.8	1.27	9.628	677
35	15-9-H	34,373.4	60.3	34,313.1	11,457.8	11,437.7	1.29	9.435	664
36	15-1-I	27,608.3	93.6	27,514.7	6,135.2	6,114.4		5.052	356
37	35-1-I	26,781.0	58.8	26,722.2	9,453.2	9,432.5		7.780	548
38	55-1-I	22,509.0	40.6	22,468.4	11,254.5	11,234.2		9.268	652
39	15-9-I	34,374.3	67.1	34,307.2	10,857.3	10,836.1		8.940	630
40	15-4-J	32,783.5	102.8	32,680.8	6,556.7	6,536.2		5.390	380
41	35-4-J	30,646.7	63.6	30,583.1	10,215.6	10,194.4		8.413	592
42	15-15-J	30,879.1	57.5	30,821.6	11,608.7	11,587.0		9.558	673
43	15-2-K	33,443.0	101.0	33,451.0	6,710.4	6,690.2		5.525	388
44	35-2-K	30,934.0	61.2	30,872.8	10,311.3	10,290.9		8.490	597

TABLE V.

BOILER PERFORMANCE—*Continued.*

COAL CONSUMPTION—BRAZIL BLOCK COAL.

Number.	Laboratory Symbol.	Total Dry Coal Fired.* Lbs.	Dry Ash Caught in Ashpan. Lbs.	Dry Cinders Caught in Smoke-box. Lbs.	Dry Coal Fired per Hour.* Lbs.	Combustible Fired per Hour by Analysis. Lbs.	Dry Coal Fired per Hour per Square Foot of Heating Surface.* Lbs.	Dry Coal Fired per Hour per Square Foot of Grate Surface.* Lbs.
1	2	32	33	34	35	36	37	38
1	15–1$_b$–V	2,577.2	261.5	31.0	813.9	732	.670	47.2
2	25–1–V	4,460.0	320.0	79.5	1,115.0	1,003	.918	64.6
3	35–1–V	3,180.0	272.0	124.5	1,362.9	1,227	1.122	78.9
4	55–1–V	3,330.0	238.5	255.0	1,665.0	1,498	1.371	96.5
5	15–1–A	3,399.0	292.0	57.0	849.7	765	.700	49.3
6	25–1–A	3,864.3	93.5	60.0	909.2	818	.748	52.8
7	35–1–A	3,784.9	299.0	135.0	1,261.6	1,135	1.038	73.1
8	45–1–A	3,272.2	231.5	135.7	1,308.9	1,178	1.078	75.9
9	55–1–A	2,980.3	219.5	162.0	1,490.2	1,341	1.227	86.4
10	15–2–A	3,296.7	257.5	71.0	1,098.9	989	.905	63.7
11	25–2–A	4,826.0	25.0	227.0	1,608.7	1,448	1.324	93.3
12	35–2–A	5,932.9	257.5	239.0	1,977.6	1,780	1.628	114.6
13	45–2–A	5,720.2	242.5	380.5	2,451.5	2,206	2.019	142.1
14	55–2–A	2,994.8	205.0	330.0	2,642.5	2,378	2.176	153.2
15	25–3–A	4,684.1	91.0	231.0	2,294.3	2,065	1.889	133.0
16	35–3–A	6,266.3	298.0	118.2	3,133.2	2,819	2.581	181.6
17	15–9–A	5,014.0	265.0	174.0	2,005.6	1,805	1.651	116.3
18	35–2–B	4,369.7	460.5	177.5	1,542.2	1,388	1.270	89.4
19	35–2–C	5,356.1	384.0	348.0	2,678.1	2,410	2.205	155.3
20	35–2–E	4,106.5	353.0	113.5	1,368.8	1,232	1.127	79.3
21	35–2–F	3,859.1	294.5	109.5	1,286.3	1,158	1.059	74.6
22	15–1–G	2,678.0	262.0	43.0	892.7	803	.735	51.8
23	35–1–G	4,288.4	341.5	141.5	1,518.1	1,362	1.246	87.7
24	35–1$_b$–G	4,150.3	380.0	179.0	1,464.8	1,318	1.206	84.9
25	55–1–G	3,768.9	241.0	314.5	1,884.5	1,696	1.552	109.2
26	35–2–G	6,182.7	363.0	168.5	2,318.5	2,087	1.909	134.4
27	35–3–G	4,707.9	365.0	157.0	2,017.7	1,816	1.662	117.0
28	15–9–G	5,486.7	354.0	168.0	2,057.5	1,852	1.694	119.3
29	15–1–H	2,374.5	240.0	37.0	791.5	713	.652	45.9
30	35–1–H	4,152.3	320.0	169.0	1,557.1	1,401	1.283	90.3
31	55–1–H	3,203.1	213.5	233.5	1,601.6	1,441	1.319	92.8
32	35–2–H	4,250.3	297.0	204.0	2,125.1	1,912	1.750	123.2
33	35–2$_b$–H	4,321.2	331.5	100.5	1,620.4	1,458	1.334	93.9
34	35–3–H	4,248.3	346.0	300.0	2,124.1	1,912	1.749	123.2
35	15–9–H	6,363.1	344.0	234.0	2,121.0	1,909	1.746	123.0
36	15–1–I							
37	35–1–I							
38	55–1–I							
39	15–9–I							
40	15–4–J							
41	35–4–J							
42	15–15–J							
43	15–2–K							
44	35–2–K							

* To obtain an approximate measure of boiler performance in terms of West Virginia or Pittsburgh coal, multiply the value of the columns 32, 35, 37. and 38 by 0.8.

TABLE VI.

BOILER PERFORMANCE—*Continued.*

Number.	Laboratory Symbol.	Quality of Steam in Dome, Moisture. Per Cent.	B. T. U.—Taken Up by Each Pound of Water.	B. T. U.—Taken Up by Boiler per Minute.	B. T. U.—Taken Up by Boiler per Pound of Coal.	B. T. U.—Taken Up by Boiler per Pound Combustible.	Actual Water Evaporated Per Pound of Coal.* Lbs.	Actual Water Evaporated Per Pound Combustible. Lbs.
1	2	39	40	41	42	43	44	45
1	15-1$_b$-V	0.63	1,162.6	108,858	8,026	8,918	6.903	7.670
2	25-1-V	0.68	1,161.8	142,688	7,678	8,531	6.609	7.343
3	35-1-V	0.63	1,164.7	165,543	7,288	8,098	6.258	6.953
4	55-1-V	0.88	1,156.3	187,109	6,743	7,492	5.831	6.479
5	15-1-A	0.49	1,164.1	107,194	7,569	8,410	6.502	7.224
6	25-1-A	0.78	1,160.0	117,798	7,771	8,638	6.701	7.434
7	35-1-A	0.83	1,161.7	157,268	7,479	8,310	6.438	7.153
8	45-1-A	0.87	1,158.1	167,336	7,671	8,523	6.623	7.379
9	55-1-A	1.22	1,154.7	172,079	6,929	7,699	6.000	6.666
10	15-2-A	0.76	1,160.2	140,778	7,686	8,541	6.625	7.361
11	25-2-A	0.90	1,158.2	185,994	6,937	7,709	5.990	6.633
12	35-2-A	1.06	1,159.3	221,072	6,707	7,452	5.786	6.429
13	45-2-A	1.24	1,155.9	240,262	5,881	6,534	5.087	5.652
14	55-2-A	1.13	1,153.9	269,996	6,130	6,812	5.313	5.903
15	25-3-A	1.11	1,158.8	249,743	6,531	7,256	5.636	6.262
16	35-3-A	1.11	1,156.3	287,871	5,513	6,126	4.768	5.296
17	15-9-A	0.94	1,157.2	220,038	6,583	7,314	5.688	6.320
18	35-2-B	0.69	1,160.5	167,624	6,521	7,245	5.619	6.243
19	35-2-C	0.70	1,166.7	252,873	5,666	6,296	4.856	5.395
20	35-2-E	0.68	1,164.1	166,188	7,283	8,093	6.258	6.953
21	35-2-F	1.00	1,165.1	157,502	7,348	8,163	6.305	7.005
22	15-1-G	1.06	1,158.6	120,340	8,088	8,988	6.981	7.756
23	35-1-G	1.14	1,156.5	180,706	7,164	7,961	6.194	6.882
24	35-1$_b$-G	1.40	1,157.4	182,072	7,458	8,287	6.444	7.160
25	55-1-G	1.31	1,156.1	200,015	6,369	7,076	5.508	6.120
26	35-2-G	1.44	1,154.6	237,025	6,134	6,815	5.313	5.903
27	35-3-G	1.62	1,154.8	223,206	6,637	7,375	5.748	6.386
28	15-9-G	1.29	1,157.0	223,243	6,510	7,234	5.627	6.252
29	15-1-H	1.62	1,152.4	116,668	8,842	9,823	7.675	8.528
30	35-1-H	1.49	1,155.3	178,516	6,879	7,643	5.954	6.615
31	55-1-H	1.29	1,158.5	198,046	7,419	8,243	6.404	7.115
32	35-2-H	1.34	1,153.5	237,746	6,713	7,459	5.819	6.465
33	35-2$_b$-H	1.20	1,160.0	187,260	6,934	7,704	5.977	6.641
34	35-3-H	1.27	1,155.9	225,266	6,366	7,071	5.505	6.116
35	15-9-H	1.29	1,156.6	220,866	6,248	6,942	5.402	6.002
36	15-1-I							
37	35-1-I							
38	55-1-I							
39	15-9-I							
40	15-4-J							
41	35-4-J							
42	15-15-J							
43	15-2-K							
44	35-2-K							

* To obtain an approximate measure of the evaporative performance of the boiler, when using West Virginia or Pittsburgh coal, multiply column 44 by 1.25.

TABLE VII.

BOILER PERFORMANCE—*Continued.*

Number.	Laboratory Symbol.	Water Evaporated per Hour.	Equivalent Evaporation from and at 212° F. Per Hour.	Per Square Foot of Heating Surface per Hour. Lbs.	Per Square Foot of Grate Surface per Hour. Lbs.	Per Pound of Dry Coal.* Lbs.	Per Pound of Combustible. Lbs.	Boiler Horsepower (34.5 Evaporative units=1B.H.P.)
1	2	46	47	48	49	50	51	52
1	15-1b-V	5,617.8	6,762	5.57	392	8.31	9.24	196
2	25-1-V	7,368.9	8,864	7.30	514	7.95	8.84	257
3	35-1-V	8,528.2	10,286	8.47	596	7.54	8.38	298
4	55-1-V	9,709.2	11,650	9.57	673	6.98	7.76	337
5	15-1-A	5,525.1	6,660	5.48	386	7.83	8.70	193
6	25-1-A	6,092.8	7,318	6.03	424	8.05	8.95	212
7	35-1-A	8,122.5	9,751	8.03	566	7.74	8.61	283
8	45-1-A	8,669.0	10,395	8.56	603	7.96	8.83	301
9	55-1-A	8,941.3	10,690	8.80	619	7.17	7.95	310
10	15-2-A	7,280.2	8,745	7.20	507	7.95	8.84	253
11	25-2-A	9,635.3	11,555	9.51	669	7.18	7.98	335
12	35-2-A	11,441.5	13,733	11.31	796	6.94	7.72	398
13	45-2-A	12,471.6	14,926	12.29	865	6.09	6.77	433
14	55-2-A	14,039.5	16,773	13.81	973	6.34	7.04	486
15	25-3-A	12,931.1	15,515	12.77	899	6.76	7.51	450
16	35-3-A	14,937.4	17,878	14.73	1,037	5.71	6.36	518
17	15-9-A	11,408.8	13,670	11.25	792	6.81	7.57	396
18	35-2-B	8,666.3	10,404	8.57	604	6.75	7.50	302
19	35-2-C	13,004.2	15,709	12.93	911	5.86	6.52	456
20	35-2-E	8,565.6	10,324	8.51	598	7.54	8.38	299
21	35-2-F	8,111.1	9,785	8.06	567	7.61	8.45	284
22	15-1-G	6,231.9	7,476	6.16	433	8.37	9.31	217
23	35-1-G	9,375.1	11,226	9.24	650	7.41	8.24	325
24	35-1b-G	9,438.8	11,312	9.31	655	7.72	8.58	328
25	55-1-G	10,380.4	12,425	10.23	720	6.59	7.33	360
26	35-2-G	12,317.3	14,725	12.12	853	6.35	7.06	427
27	35-3-G	11,597.2	13,866	11.42	804	6.87	7.63	402
28	15-9-G	11,577.1	13,869	11.42	804	6.74	7.49	402
29	15-1-H	6,074.5	7,248	5.97	420	9.15	10.17	210
30	35-1-H	9,271.2	11,090	9.13	643	7.12	7.92	321
31	55-1-H	10,256.6	12,302	10.13	713	7.68	8.54	356
32	35-2-H	12,366.6	14,769	12.16	856	6.95	7.73	428
33	35-2b-H	9,685.9	11,633	9.58	674	7.18	7.97	337
34	35-3-H	11,692.9	13,993	11.52	811	6.59	7.32	406
35	15-9-H	11,457.8	13,752	11.32	796	6.47	7.18	398
36	15-1-I	6,135.2						
37	35-1-I	9,453.2						
38	55-1-I	11,254.5						
39	15-9-I	10,857.3						
40	15-4-J	6,556.7						
41	35-4-J	10,215.6						
42	15-15-J	11,608.7						
43	15-2-K	6,710.4						
44	35-2-K	10,311.3						

* To obtain an approximate measure of the evaporative performance of the boiler, when using West Virginia or Pittsburgh coal, multiply column 50 by 1.25.

TABLE VIII.
ENGINE PERFORMANCE.
EVENTS OF STROKE BY INDICATOR.

Number.	Laboratory Symbol.	Admission Per Cent of Stroke. Right Side. H.E.	Right Side. C.E.	Left Side. H.E.	Left Side. C.E.	Average.	Cut-off Per Cent of Stroke. Right Side. H.E.	Right Side. C.E.	Left Side. H.E.	Left Side. C.E.	Average.
1	2	53	54	55	56	57	58	59	60	61	62
1	15–1$_b$–V	3.05	6.10	4.60	6.60	5.09	24.50	24.80	25.00	26.00	25.08
2	25–1–V	3.05	6.10	4.60	6.60	5.09	24.50	24.80	25.00	26.00	25.08
3	35–1–V	3.05	6.10	4.60	6.60	5.09	24.50	24.80	25.00	26.00	25.08
4	55–1–V	Car	ds un	satisf	actory		26.04	25.14	26.09	23.95	25.31
5	15–1–A	2.25	4.25	3.50	3.00	3.25	23.50	25.25	25.75	24.25	24.68
6	25–1–A	2.25	4.25	3.50	3.00	3.25	23.50	25.25	25.75	24.25	24.68
7	35–1–A	2.25	4.25	3.50	3.00	3.25	23.50	25.25	25.75	24.25	24.68
8	45–1–A	1.79	2.64	1.79	2.39	2.15	21.36	23.04	22.97	22.36	22.43
9	55–1–A	Car	ds un	satisf	actory		23.16	23.54	24.98	22.53	22.61
10	15–2–A	1.10	1.70	1.40	1.80	1.50	39.10	36.30	38.80	37.10	37.80
11	25–2–A	1.09	1.86	1.28	1.91	1.53	33.26	34.61	34.92	34.69	34.37
12	35–2–A	1.00	2.50	2.00	1.30	1.70	32.75	33.00	35.25	33.75	33.69
13	45–2–A	1.39	2.40	1.11	2.10	1.75	32.40	33.00	33.80	33.10	33.07
14	55–2–A	Car	ds un	satisf	actory		35.30	34.70	35.60	36.00	35.40
15	25–3–A	1.20	1.90	1.00	1.20	1.32	44.40	43.70	44.80	44.00	44.22
16	35–3–A	1.81	1.69	0.46	1.08	1.51	43.71	44.97	43.98	42.63	43.82
17	15–9–A	0.00	0.00	0.00	0.00	0.00	80.16	79.88	81.30	79.93	80.32
18	35–2–B	1.58	1.77	1.55	1.66	1.64	29.30	31.19	35.13	30.75	31.59
19	35–2–C	1.28	.06	1.21	1.75	1.57	29.06	31.23	35.55	31.70	31.88
20	35–2–E	1.61	.08	1.67	1.79	1.79	29.01	30.73	29.03	27.13	28.97
21	35–2–F	1.83	2.21	1.83	1.83	1.92	28.20	30.23	25.33	28.14	27.97
22	15–1–G	Car	ds un	satisf	actory		25.69	26.06	26.15	25.32	25.80
23	35–1–G	Car	ds un	satisf	actory		26.89	25.31	27.46	25.72	26.34
24	35–1$_b$–G	4.42	6.47	3.72	4.95	4.89	25.94	24.79	26.84	24.53	25.52
25	55–1–G	6.99	6.51	4.95	5.09	5.89	25.58	24.62	25.36	23.16	24.68
26	35–2–G	3.18	4.12	3.20	3.83	3.58	35.29	35.36	33.77	34.30	34.68
27	35–3–G	2.25	2.56	2.24	2.18	2.31	45.39	44.18	45.89	45.04	45.12
28	15–9–G	.95	1.04	1.10	1.12	1.05	81.07	79.83	80.91	79.69	80.38
29	15–1–H	2.35	3.77	2.64	3.19	2.99	26.93	25.90	26.64	24.23	25.92
30	35–1–H	3.24	5.17	3.31	3.74	3.86	24.52	24.80	27.60	24.21	25.28
31	55–1–H	3.21	3.29	2.84	3.02	3.09	25.33	24.58	25.82	24.10	24.96
32	35–2–H	Car	ds un	satisf	actory		35.00	36.42	37.35	36.23	36.25
33	35–2$_b$–H	2.77	2.78	2.47	2.22	2.56	35.94	34.45	37.09	35.81	35.83
34	35–3–H	0.32	2.00	1.07	1.77	1.54	42.51	42.53	45.50	47.25	44.45
35	15–9–H	Car	ds un	satisf	actory		83.92	82.87	83.90	87.67	83 35
36	15–1–I	2.20	3.61	1.35	3.35	2.63	23.72	26.34	24.05	25.85	24.99
37	35–1–I	2.50	4.58	1.44	3.29	2.95	20.85	22.61	19.38	21.11	20.99
38	55–1–I	3.16	3.65	2.00	2.93	2.94	17.33	22.10	18.37	19.76	19.39
39	15–9–I	0.00	0.00	0.00	0.00	0.00	82.34	81.77	83.03	82.97	82.53
40	15–4–J	1.27	1.71	1.41	1.65	1.51	26.70	24.03	27.63	23.49	25.46
41	35–4–J	1.39	1.58	0.63	1.51	1.53	23.90	21.92	25.06	21.96	23.21
42	15–15–J	0.00	0.00	0.00	0.00	0.00	72.26	68.31	70.65	67.85	69.76
43	15–2–K	2.10	2.68	4.70	4.70	3.55	29.30	27.51	29.61	27.11	28.38
44	35–2–K	1.99	2.75	4.14	4.33	3.30	28.30	28.22	30.86	26.75	28.53

Table IX.

ENGINE PERFORMANCE—*Continued.*

Events of Stroke by Indicator.

Number.	Laboratory Symbol.	Release Per Cent of Stroke.					Compression Per Cent of Stroke.				
		Right Side.		Left Side.		Average.	Right Side.		Left Side.		Average.
		H.E.	C.E.	H.E.	C.E.		H.E.	C.E.	H.E.	C.E.	
1	2	63	64	65	66	67	68	69	70	71	72
1	15-1$_b$-V	65.50	70.00	66.00	69.10	67.65	38.00	43.50	42.00	45.00	42.125
2	25-1-V	65.50	70.00	66.00	69.10	67.65	38.00	43.50	42.00	45.00	42.125
3	35-1-V	65.50	70.00	66.00	69.10	67.65	38.00	43.50	42.00	45.00	42.125
4	55-1-V	72.90	68.27	73.31	69.35	70.96	21.96	23.53	22.37	23.75	22.90
5	15-1-A	37.80	72.00	73.00	71.50	71.08	31.00	33.00	35.50	34.00	33.37
6	25-1-A	67.80	72.00	73.00	71.50	71.08	31.00	33.00	35.50	34.00	33.37
7	35-1-A	67.80	72.00	73.00	71.50	71.08	31.00	33.00	35.50	34.00	33.37
8	45-1-A	73.07	73.04	75.07	72.18	73.34	21.89	24.11	20.82	23.96	22.69
9	55-1-A	76.16	74.13	73.35	73.12	74.20	22.56	23.79	21.15	22.96	22.62
10	15-2-A	75.60	75.10	76.00	76.60	75.80	23.50	23.70	20.30	19.40	21.72
11	25-2-A	74.94	72.61	76.00	74.50	74.51	28.12	27.11	26.31	28.44	27.49
12	35-2-A	77.75	78.50	78.00	77.50	77.94	27.75	28.50	28.75	29.50	28.62
13	45-2-A	77.70	78.10	79.20	78.00	78.25	22.60	23.90	25.60	25.90	24.50
14	55-2-A	76.90	77.50	77.60	75.60	76.90	27.40	27.20	27.40	27.30	27.32
15	25-3-A	78.40	79.30	80.90	79.80	79.60	15.80	17.50	16.00	17.20	16.62
16	35-3-A	81.00	78.75	82.06	80.37	80.54	23.29	25.85	23.62	25.21	24.49
17	15-9-A	93.54	93.75	94.10	93.04	93.61	7.82	6.57	8.90	7.54	7.71
18	35-2-B	77.75	80.13	77.30	81.65	79.21	29.38	30.66	29.72	30.22	29.99
19	35-2-C	77.58	78.33	78.06	78.95	78.23	30.78	30.78	25.03	28.20	28.69
20	35-2-E	78.70	79.98	77.35	79.30	78.83	30.10	31.40	28.45	28.95	29.72
21	35-2-F	77.21	78.00	77.69	78.05	77.74	29.93	29.45	30.10	28.45	29.48
22	15-1-G	63.10	61.14	64.32	63.00	62.89	31.67	28.18	31.24	30.29	30.34
23	35-1-G	63.88	62.12	65.90	63.38	63.82	35.15	33.81	33.31	32.03	33.57
24	35-1$_b$-G	64.12	63.09	65.11	64.08	64.10	33.63	35.71	33.38	34.14	34.21
25	55-1-G	65.53	65.73	65.77	66.61	65.91	34.73	37.18	33.91	35.94	35.44
26	35-2-G	69.89	70.28	72.37	68.63	70.29	26.25	27.37	26.90	25.40	26.48
27	35-3-G	76.00	75.61	76.89	74.93	75.86	26.04	23.57	24.57	22.93	24.28
28	15-9-G	92.39	92.77	92.30	91.21	92.17	6.84	5.45	7.49	6.39	6.54
29	15-1-H	59.16	59.00	60.04	57.74	58.98	23.67	24.66	24.04	23.25	23.90
30	35-1-H	60.33	57.64	61.84	58.18	59.50	27.99	27.86	28.70	26.88	27.86
31	55-1-H	62.08	59.67	58.95	59.35	60.01	23.44	21.37	20.43	21.65	21.72
32	35-2-H	66.29	67.78	69.10	68.00	67.79	25.29	25.80	24.35	24.73	25.04
33	35-2$_b$-H	65.42	65.27	68.16	65.53	66.09	24.87	23.70	23.78	22.97	23.83
34	35-3-H	71.41	71.65	73.42	71.90	72.10	21.34	21.20	16.44	19.40	19.60
35	15-9-H	92.46	91.47	92.69	91.54	92.04	6.86	6.28	6.65	6.40	6.55
36	15-1-I	47.02	50.82	48.61	51.22	49.42	15.74	19.82	14.60	19.14	17.325
37	35-1-I	49.95	53.05	49.79	52.28	51.27	16.64	19.94	15.00	18.86	17.61
38	55-1-I	53.80	55.33	53.50	50.50	53.28	19.46	19.40	17.87	19.55	19.07
39	15-9-I	89.84	89.08	90.50	89.92	89.84	9.66	9.03	8.76	8.95	9.10
40	15-4-J	71.53	68.74	70.78	67.48	69.63	33.35	31.46	34.48	31.95	32.81
41	35-4-J	71.35	68.21	70.50	69.56	69.90	34.03	31.10	32.68	30.99	32.20
42	15-15-J	91.62	89.25	90.65	89.53	90.26	9.41	8.64	11.01	9.80	9.71
43	15-2-K	70.76	68.70	67.15	64.86	67.87	31.86	28.85	35.58	32.66	32.24
44	35-2-K	72.47	71.30	70.08	68.41	70.57	32.22	32.03	34.47	31.64	32.59

TABLE X.

ENGINE PERFORMANCE—*Continued.*

PRESSURES ABOVE ATMOSPHERE BY INDICATOR.

Number.	Laboratory Symbol.	Initial Pressure, Pounds per Square Inch.					Pressure at Cut-off, Pounds per Square Inch.				
		Right Side.		Left Side.		Average.	Right Side.		Left Side.		Average.
		H.E.	C.E.	H.E.	C.E.		H.E.	C.E.	H.E.	C.E.	
1	2	73	74	75	76	77	78	79	80	81	82
1	15–1$_b$–V	123.10	126.00	126.00	128.65	125.94	92.11	93.68	103.10	93.41	95.57
2	25–1–V	127.90	132.80	127.20	135.00	130.72	83.10	85.40	88.60	86.90	86.00
3	35–1–V	111.78	115.00	116.70	117.30	115.20	69.80	78.14	75.64	78.80	75.59
4	55–1–V	118.17	123.45	119.12	118.09	119.71	57.12	67.64	57.25	65.95	61.99
5	15–1–A	118.20	120.54	122.88	122.60	121.06	89.11	91.75	91.29	91.81	90.99
6	25–1–A	101.48	100.58	104.42	105.58	103.02	67.04	68.88	70.25	72.46	69.66
7	35–1–A	121.80	120.60	128.10	130.00	125.13	67.33	71.20	70.80	73.10	70.61
8	45–1–A	111.40	111.10	118.00	117.50	114.50	64.70	70.70	64.60	71.60	67.90
9	55–1–A	116.93	120.82	119.20	119.64	119.15	57.43	63.75	54.30	63.86	59.83
10	15–2 A	115.30	122.30	121.60	123.60	120.70	85.20	90.90	84.90	90.80	87.90
11	25–2–A	118.40	117.40	121.50	122.80	120.02	79.60	81.50	83.20	84.20	82.12
12	35–2–A	109.38	119.90	114.94	118.17	115.60	68.65	78.00	72.66	74.19	73.37
13	45–2–A	111.10	116.90	118.60	117.30	115.97	60.10	68.20	60.10	67.20	63.90
14	55–2–A	118.00	118.30	118.10	117.40	117.95	51.60	59.30	52.70	56.40	55.07
15	25–3–A	113.30	119.70	120.80	120.90	118.67	80.00	85.40	84.10	85.60	83.70
16	35–3–A	110.88	122.29	117.04	115.83	116.51	71.08	75.58	73.68	78.66	74.75
17	15–9–A	74.80	75.00	75.13	75.76	75.17	68.03	68.16	70.73	70.33	69.30
18	35–2–B	91.25	93.53	88.19	93.22	91.55	55.75	58.39	53.00	58.47	56.40
19	35–2–C	134.00	140.75	134.46	140.00	137.30	89.67	92.67	86.83	90.75	89.98
20	35–2–E	87.59	89.94	87.44	88.72	88.42	58.82	59.18	62.38	61.72	60.52
21	35–2–F	86.72	87.95	85.67	88.20	87.16	58.78	59.56	65.50	61.00	61.21
22	15–1–G	119.86	119.72	123.30	124.03	121.98	92.94	96.64	98.17	96.75	96.12
23	35–1–G	106.64	107.87	112.56	112.57	109.91	68.44	73.28	71.38	70.50	70.90
24	35–1$_b$–G	110.67	111.06	116.44	118.37	114.14	71.52	74.44	71.53	75.81	74.08
25	55–1–G	99.50	102.75	113.00	114.30	107.38	60.50	65.00	65.54	68.50	64.88
26	35–2–G	116.35	113.09	119.53	123.40	118.09	69.96	73.64	77.06	77.23	74.47
27	35–3–G	81.21	83.00	83.75	82.43	82.60	48.93	52.18	52.00	51.25	51.09
28	15–9–G	80.00	78.37	78.75	80.00	79.28	70.03	69.96	72.66	72.66	71.33
29	15–1–H	118.36	119.78	123.44	123.53	121.28	89.14	93.54	90.62	93.78	91.77
30	35–1–H	107.81	106.53	114.90	115.21	111.11	66.63	72.35	65.33	69.50	68.45
31	55–1–H	Cards	unsati	sfactor	y		56.42	60.58	57.27	59.80	58.52
32	35–2–H	106.21	103.87	108.41	110.08	107.14	60.96	62.29	60.75	60.58	61.14
33	35–2$_b$–H	84.41	85.47	88.45	87.91	86.56	47.16	50.69	49.31	47.97	48.78
34	35–3–H	71.63	75.09	75.09	73.25	73.76	45.45	49.91	46.04	45.50	46.72
35	15–9–H	76.21	74.42	75.00	76.44	75.52	64.92	65.36	68.50	68.22	66.75
36	15–1–I	114.51	120.04	118.52	123.22	119.07	88.19	94.27	92.96	93.29	92.18
37	35–1–I	110.65	109.76	115.47	119.35	113.81	74.83	83.35	79.12	82.76	79.87
38	55–1–I	98.91	114.91	111.25	115.58	110.16	69.50	69.41	71.91	73.41	71.06
39	15–9–I	71.72	69.72	69.42	72.16	70.76	63.52	65.61	65.31	64.58	64.76
40	15–4–J	119.77	126.15	123.91	124.50	123.58	93.92	96.97	97.28	99.33	96.82
41	35–4–J	119.16	122.08	127.91	128.91	124.52	80.86	85.58	85.86	87.00	84.83
42	15–15–J	88.59	88.18	88.28	90.59	88.91	78.46	78.75	79.09	82.00	79.58
43	15–2–K	119.42	122.97	122.92	123.13	122.11	91.70	92.47	97.83	96.87	94.72
44	35–2–K	115.50	117.19	113.25	120.36	116.57	71.75	72.03	74.78	78.86	74.36

TABLE XI.

ENGINE PERFORMANCE—*Continued.*

PRESSURES ABOVE ATMOSPHERE BY INDICATOR.

Number.	Laboratory Symbol.	Pressure at Release, Pounds per Square Inch.					Pressure at Beginning of Compression, Pounds per Square Inch.				
		Right Side.		Left Side.		Average.	Right Side.		Left Side.		Average.
		H.E.	C.E.	H.E.	C.E.		H.E.	C.E.	H.E.	C.E.	
1	2	83	84	85	86	87	88	89	90	91	92
1	15-1b-V	32.50	29.53	35.42	30.76	32.05	3.47	2.95	2.66	2.15	2.81
2	25-1-V	28.90	28.06	31.50	27.90	29.09	6.33	3.98	3.94	3.46	4.43
3	35-1-V	23.14	24.50	26.14	23.70	24.37	7.78	8.57	7.50	7.57	7.86
4	55-1-V	15.96	20.64	14.62	18.54	17.44	31.21	30.23	27.12	27.04	28.90
5	15-1-A	29.16	28.79	28.02	27.63	28.40	8.05	8.71	4.58	4.38	6.43
6	25-1-A	20.40	19.33	19.67	20.33	19.93	10.40	9.15	6.00	6.96	8.13
7	35-1-A	20.47	20.20	20.30	20.80	20.44	12.25	11.60	9.30	10.50	10.91
8	45-1-A	14.90	17.30	14.60	17.50	16.10	22.60	23.00	24.90	23.10	23.40
9	55-1-A	13.52	15.45	13.18	15.32	14.37	24.69	27.73	27.30	28.18	26.98
10	15-2-A	38.30	42.80	38.60	39.30	39.70	4.60	7.60	8.60	9.30	7.50
11	25-2-A	32.70	36.70	33.20	34.03	34.16	6.00	7.90	6.80	6.50	6.80
12	35-2-A	25.05	29.00	29.02	27.19	27.57	9.50	12.03	10.90	11.50	10.98
13	45-2-A	21.00	23.70	21.30	24.00	22.50	17.25	21.10	16.10	18.80	18.31
14	55-2-A	19.60	23.00	19.40	22.30	21.07	17.30	21.20	17.70	20.40	19.15
15	25-3-A	40.50	44.40	41.30	42.40	42.15	13.40	13.40	12.30	11.80	12.72
16	35-3-A	34.25	39.95	35.36	38.41	36.99	13.75	16.58	15.12	15.66	15.28
17	15-9-A	56.10	55.73	58.43	57.73	57.00	1.80	2.00	1.77	1.73	1.82
18	35-2-B	18.25	18.53	19.56	17.72	18.51	9.06	9.30	7.81	9.34	8.88
19	35-2-C	31.17	34.21	34.88	32.89	33.28	11.79	12.83	10.08	12.42	11.78
20	35-2-E	17.68	18.12	20.00	17.28	18.27	9.30	9.09	9.56	9.67	9.40
21	35-2-F	18.06	18.89	18.67	18.47	18.52	8.28	9.17	7.67	9.53	8.66
22	15-1-G	36.44	42.83	37.44	35.69	38.10	4.22	6.94	5.25	4.88	5.32
23	35-1-G	24.85	27.72	26.76	24.86	25.05	7.68	9.25	9.47	9.13	8.88
24	35-1b-G	25.59	28.33	26.59	24.97	26.37	9.53	8.73	9.03	7.87	8.79
25	55-1-G	19.91	20.33	20.64	19.20	20.02	14.33	12.42	14.10	12.20	13.26
26	35-2-G	32.25	33.90	33.20	34.13	33.37	11.14	11.25	9.90	10.86	10.78
27	35-3-G	25.93	26.93	27.61	26.78	26.81	11.82	11.36	10.43	10.18	10.95
28	15-9-G	59.16	57.70	61.00	60.28	59.53	2.50	2.67	2.22	2.00	2.35
29	15-1-H	39.50	43.28	38.31	36.91	39.50	12.78	11.41	10.15	10.20	11.13
30	35-1-H	24.42	29.54	26.37	25.75	26.52	10.69	12.04	9.40	9.75	10.47
31	55-1-H	19.33	22.33	22.09	20.20	20.99	20.92	27.25	23.45	21.10	23.18
32	35-2-H	29.25	31.04	30.00	28.71	29.75	9.21	9.66	9.00	8.54	9.10
33	35-2b-H	22.25	24.03	23.53	22.75	23.14	9.34	9.97	9.37	8.34	9.25
34	35-3-H	23.86	26.64	24.86	26.33	25.42	7.90	8.68	10.00	8.42	8.75
35	15-9-H	57.03	57.25	59.41	59.19	58.22	2.03	2.31	2.47	1.50	2.08
36	15-1-I	44.70	51.54	45.93	44.41	46.64	19.70	14.08	15.48	13.00	15.57
37	35-1-I	30.65	34.71	31.18	31.59	32.03	21.76	18.88	19.47	16.35	19.12
38	55-1-I	21.75	26.16	22.66	26.83	24.35	20.41	22.75	19.50	19.00	20.42
39	15-9-I	56.47	58.66	58.00	57.32	57.61	2.18	00.43	1.50	1.26	1.34
40	15-4-J	32.55	36.10	33.31	31.32	33.32	2.10	2.08	2.73	2.51	2.35
41	35-4-J	24.06	25.69	26.85	23.38	24.99	8.03	9.31	8.05	7.35	8.18
42	15-15-J	56.84	57.28	57.68	56.46	57.07	2.71	1.62	2.25	1.81	2.09
43	15-2-K	36.32	37.82	39.60	37.71	37.86	3.96	4.65	3.80	3.83	4.06
44	35-2-K	24.37	25.53	29.03	25.80	26.18	10.19	10.97	9.22	9.78	10.04

TABLE XII.

ENGINE PERFORMANCE—*Continued.*

PRESSURES ABOVE ATMOSPHERE BY INDICATOR.

Number.	Laboratory Symbol.	Least Back Pressure, Pounds per Square Inch. Right Side. H.E.	C.E.	Left Side. H.E.	C.E.	Average.	Mean Effective Pressure, Pounds per Square Inch. Right Side. H.E.	C.E.	Left Side. H.E.	C.E.	Average.
1	2	93	94	95	96	97	98	99	100	101	102
1	15-1b-V	2.18	1.53	1.22	0.92	1.46	41.04	41.97	47.60	45.12	43.93
2	25-1-V	2.65	2.23	1.88	1.19	1.94	35.35	36.98	43.28	39.31	38.73
3	35-1-V	3.53	3.03	2.90	2.18	2.91	27.27	28.68	35.12	32.29	30.84
4	55-1-V	3.21	4.68	2.12	3.28	3.32	17.29	20.67	19.54	20.19	19.42
5	15-1-A	1.62	1.60	1.06	0.94	1.31	39.95	42.83	46.44	44.76	43.50
6	25-1-A	1.80	2.22	1.60	1.43	1.76	27.05	30.08	33.86	32.32	30.83
7	35-1-A	3.70	3.05	2.50	2.77	3.01	26.01	28.40	31.08	30.73	29.06
8	45-1-A	2.20	2.90	2.70	2.80	2.70	20.90	22.60	22.30	23.80	22.40
9	55-1-A	2.70	3.39	2.25	2.90	2.81	16.63	17.61	17.69	18.33	17.56
10	15-2-A	0.90	0.70	1.20	0.40	0.80	60.93	62.93	62.96	64.32	62.78
11	25-2-A	3.00	2.90	2.47	2.90	2.82	49.12	52.09	53.98	53.85	52.26
12	35-2-A	5.05	6.40	5.16	5.36	5.49	39.38	42.65	44.44	42.21	42.17
13	45-2-A	5.07	5.90	5.10	5.60	5.47	30.69	33.52	32.94	34.14	32.82
14	55-2-A	4.90	6.75	5.10	6.60	5.84	25.33	26.67	26.55	27.10	26.41
15	25-3-A	3.50	3.90	3.40	3.30	3.52	61.18	64.40	66.34	66.14	64.51
16	35-3-A	8.33	10.08	7.91	10.09	9.10	47.86	49.19	50.73	51.14	49.73
17	15-9-A	1.03	1.75	1.30	0.64	1.18	66.68	66.39	68.68	68.98	67.68
18	35-2-B	3.06	3.41	2.93	2.88	3.07	27.59	28.89	31.13	29.88	29.37
19	35-2-C	6.79	7.41	7.79	6.75	7.19	47.92	49.85	55.13	50.27	50.79
20	35-2-E	3.24	3.35	3.47	2.89	3.24	27.82	28.64	31.16	28.63	29.06
21	35-2-F	2.89	2.94	2.46	3.13	2.86	28.33	29.54	30.59	28.83	29.32
22	15-1-G	0.36	1.80	0.80	0.14	0.78	47.48	48.07	49.93	47.69	48.39
23	35-1-G	2.91	2.06	2.15	1.43	2.14	29.71	29.92	33.88	32.23	31.43
24	35-1b-G	3.00	2.53	1.56	1.87	2.24	29.75	31.32	33.52	32.69	31.82
25	55-1-G	2.92	3.29	2.95	2.85	3.00	19.63	21.21	23.29	22.72	21.71
26	35-2-G	3.96	5.47	4.00	4.56	4.50	42.89	43.07	47.08	46.66	44.92
27	35-3-G	6.61	5.11	4.50	4.46	5.17	31.47	34.54	37.15	36.32	34.87
28	15-9-G	0.69	1.43	0.50	0.12	0.69	69.00	69.00	70.41	71.67	70.02
29	15-1-H	0.70	2.23	0.69	0.48	1.01	45.08	46.11	46.55	44.83	45.64
30	35-1-H	1.23	3.75	1.80	1.37	2.04	29.22	30.93	33.57	31.88	31.40
31	55-1-H	3.04	3.29	2.18	2.10	2.65	20.36	19.92	23.55	22.49	21.58
32	35-2-H	3.21	5.33	3.33	3.33	3.80	37.57	38.17	40.82	40.31	39.22
33	35-2b-H	2.59	2.42	2.34	2.10	2.36	28.17	29.92	31.90	31.20	30.30
34	35-3-H	3.31	4.31	2.81	3.17	3.40	31.53	33.07	35.27	35.72	33.90
35	15-9-H	0.79	1.54	1.06	0.46	0.96	65.52	65.32	67.44	68.50	66.70
36	15-1-I	1.28	0.17	0.76	0.61	0.71	37.30	44.92	42.17	44.94	42.33
37	35-1-I	2.82	1.97	1.29	1.14	1.81	25.98	33.33	30.71	34.30	31.08
38	55-1-I	4.00	3.66	2.50	3.16	3.33	20.49	24.15	24.37	25.72	23.68
39	15-9-I	2.10	0.03	1.42	0.92	1.09	59.80	62.24	61.76	63.04	61.71
40	15-4-J	0.63	0.68	1.19	0.36	0.72	50.84	50.13	52.78	49.58	50.83
41	35-4-J	3.00	3.47	3.08	2.08	2.91	34.55	34.94	40.11	37.75	36.84
42	15-15-J	1.71	0.75	0.94	0.70	1.03	73.24	74.10	74.95	76.86	74.79
43	15-2-K	0.92	0.57	1.08	1.00	0.89	52.63	51.80	53.09	50.05	51.89
44	35-2-K	3.14	3.88	3.25	2.28	3.14	34.17	33.78	40.06	38.03	36.51

TABLE XIII.

ENGINE PERFORMANCE—*Continued.*

WEIGHT OF STEAM SHOWN BY INDICATOR-CARDS.

Number.	Laboratory Symbol.	Pounds Steam at Cut-off.					Pounds Steam at Release.				
		Right Side.		Left Side.		Total.	Right Side.		Left Side.		Total.
		H.E.	C.E.	H.E.	C.E.		H.E.	C.E.	H.E.	C.E.	
1	2	103	104	105	106	107	108	109	110	111	112
1	15–1$_b$–V	.26352	.25402	.29059	.26872	1.07685	.26819	.25583	.28372	.26187	1.06961
2	25–1–V	.24217	.23532	.25616	.25309	0.98673	.24776	.24565	.26182	.24518	1.00041
3	35–1–V	.21107	.21924	.22629	.23399	0.89059	.21660	.22724	.23372	.22223	0.89978
4	55–1–V	.18443	.20335	.18946	.19103	0.76827	.19283	.20507	.18726	.19421	0.77937
5	15–1–A	.24892	.25295	.26866	.25199	1.02252	.25725	.25780	.26627	.25177	1.03309
6	25–1–A	.19828	.20173	.21841	.20891	0.82733	.20895	.20489	.21728	.21099	0.84211
7	35–1–A	.19993	.20723	.21998	.21062	0.83776	.20969	.21039	.22162	.21423	0.85593
8	45–1–A	.17548	.19777	.18948	.19392	0.75665	.18643	.19673	.19062	.19471	0.76850
9	55–1–A	.16958	.18517	.17613	.17824	0.70912	.18418	.18764	.17742	.18355	0.73279
10	15–2–A	.34479	.33852	.34738	.34153	1.37223	.33426	.35169	.34023	.33524	1.36142
11	25–2–A	.28702	.29871	.31464	.30476	1.20513	.29843	.30696	.30750	.29677	1.20966
12	35–2–A	.25858	.27234	.28397	.27162	1.08651	.26542	.28001	.28875	.26782	1.10200
13	45–2–A	.22603	.25015	.23811	.24589	0.96018	.23531	.24814	.24305	.24827	0.97477
14	55–2–A	.21642	.23419	.22518	.23040	0.90619	.22533	.24305	.22760	.23225	0.92823
15	25–3–A	.36433	.37349	.38739	.37369	1.49890	.35918	.37890	.37711	.36703	1.48222
16	35–3–A	.32731	.34660	.34338	.34007	1.35736	.32928	.34914	.34288	.34416	1.36546
17	15–9–A	.53454	.52264	.56200	.53284	2.15202	.53132	.51815	.55392	.52534	2.12873
18	35–2–B	.20289	.20840	.22277	.21062	0.84468	.22118	.21961	.22706	.21967	0.88752
19	35–2–C	.29154	.29929	.32882	.30377	1.22342	.30198	.31095	.32490	.30679	1.24462
20	35–2–E	.20986	.20832	.21765	.20008	0.83591	.22054	.21729	.23056	.21191	0.88030
21	35–2–F	.20589	.20719	.20484	.20419	0.82211	.22042	.21833	.22427	.21748	0.88050
22	15–1–G	.26699	.27713	.28967	.26941	1.10320	.27554	.29416	.28809	.26464	1.12243
23	35–1–G	.21709	.21799	.23313	.21207	0.88028	.21940	.22444	.23783	.21264	0.89431
24	35–1$_b$–G	.21879	.21743	.22966	.21698	0.88286	.22404	.23067	.23452	.21528	0.90451
25	55–1–G	.19028	.19452	.20578	.19230	0.78288	.19738	.19641	.20383	.19147	0.78909
26	35–2–G	.27234	.28099	.28904	.28264	1.12501	.27903	.28393	.29562	.27764	1.13622
27	35–3–G	.25516	.25782	.27295	.25677	1.04270	.26112	.26055	.27624	.25613	1.05404
28	15–9–G	.55337	.53405	.57241	.54601	2.20584	.54767	.52771	.56316	.53403	2.17257
29	15–1–H	.26712	.26851	.27492	.25433	1.06488	.27474	.28710	.27541	.25097	1.08822
30	35–1–H	.19869	.21268	.21850	.20094	0.83081	.20666	.21909	.22327	.20182	0.85084
31	55–1–H	.17910	.18410	.18812	.17819	0.72951	.18491	.19009	.19235	.17779	0.74514
32	35–2–H	.24364	.25282	.26043	.24477	1.00160	.25035	.26006	.26621	.24681	1.02343
33	35–2$_b$–H	.20597	.20779	.22197	.20418	0.83991	.21008	.21504	.22689	.20791	0.85992
34	35–3–H	.22899	.24185	.24796	.24487	0.96367	.23484	.24664	.24857	.24418	0.97423
35	15–9–H	.53796	.52304	.56394	.53606	2.16100	.53232	.51711	.55336	.52769	2.13048
36	15–1–I	.24169	.27401	.26128	.26575	1.04272	.24700	.28755	.26234	.25827	1.05515
37	35–1–I	.19346	.22260	.19810	.20822	0.82238	.20145	.22598	.20555	.20868	0.84167
38	55–1–I	.16121	.18978	.17755	.18210	0.71064	.17473	.19599	.17111	.18308	0.72492
39	15–9–I	.51982	.51815	.53847	.51591	2.09236	.51459	.51416	.53161	.50683	2.06720
40	15–4–J	.27682	.26256	.29949	.26118	1.10005	.28630	.29009	.29032	.25849	1.12520
41	35–4–J	.22652	.22238	.22927	.22366	0.90183	.23650	.23179	.25199	.22170	0.94199
42	15–15–J	.54516	.51025	.53423	.52072	2.11037	.52687	.50627	.53081	.49947	2.06341
43	15–2–K	.31595	.28291	.29104	.27764	1.16754	.30944	.28079	.30300	.29747	1.19070
44	35–2–K	.26119	.23660	.23306	.23175	0.96261	.26092	.22987	.24006	.23853	0.96938

TABLE XIV.

ENGINE PERFORMANCE—*Continued.*

WEIGHT OF STEAM AS SHOWN BY INDICATOR-CARDS.

Number.	Laboratory Symbol.	Pounds of Steam at Compression. Right Side. H.E.	Right Side. C.E.	Left Side. H.E.	Left Side. C.E.	Total.
1	2	113	114	115	116	117
1	15-1b-V	.06911	.07131	.07116	.07116	.28274
2	25-1-V	.07870	.07452	.07523	.07564	.30409
3	35-1-V	.08383	.09194	.08910	.09195	.35682
4	55-1-V	.10704	.10986	.10201	.10224	.42115
5	15-1-A	.07287	.07444	.06854	.06388	.27973
6	25-1-A	.08022	.07587	.07351	.07228	.30182
7	35-1-A	.08611	.08363	.08501	.08378	.33853
8	45-1-A	.08761	.09453	.09211	.09352	.36778
9	55-1-A	.09399	.10440	.09815	.10197	.39851
10	15-2-A	.04931	.05679	.05483	.05266	.21359
11	25-2-A	.06009	.06323	.06070	.06109	.24511
12	35-2-A	.07127	.07550	.07653	.07768	.30098
13	45-2-A	.07733	.08941	.08373	.08814	.33861
14	55-2-A	.08954	.09873	.09269	.09610	.37706
15	25-3-A	.05408	.05797	.05433	.05368	.22006
16	35-3-A	.07087	.08327	.07682	.07882	.30978
17	15-9-A	.02194	.02112	.02448	.02171	.08925
18	35-2-B	.07276	.07179	.06925	.07261	.20641
19	35-2-C	.08366	.08216	.06689	.07746	.31016
20	35-2-E	.07510	.07280	.07217	.07152	.29158
21	35-2-F	.07232	.07006	.07036	.07086	.28360
22	15-1-G	.06049	.06250	.06426	.05946	.24671
23	35-1-G	.07763	.07953	.08161	.07535	.31412
24	35-1b-G	.08090	.08128	.08036	.07518	.31772
25	55-1-G	.09815	.09605	.09739	.09210	.38369
26	35-2-G	.07106	.07318	.07069	.06775	.28268
27	35-3-G	.07187	.06553	.06708	.06095	.26543
28	15-9-G	.02169	.02063	.02346	.02078	.08656
29	15-1-H	.06931	.06772	.06530	.06155	.26388
30	35-1-H	.07340	.07636	.07280	.06776	.29032
31	55-1-H	.08827	.09637	.08798	.08295	.35557
32	35-2-H	.06434	.06613	.06364	.06083	.25494
33	35-2b-H	.06379	.06292	.06341	.05717	.24729
34	35-3-H	.05333	.05479	.05042	.05060	.20914
35	15-9-H	.02101	.02116	.02350	.02006	.08573
36	15-1-I	.06643	.06442	.05723	.06014	.24721
37	35-1-I	.07150	.07411	.06528	.06614	.27703
38	55-1-I	.07658	.07232	.07285	.07327	.29501
39	15-9-I	.02489	.02207	.02393	.02284	.09373
40	15-4-J	.05652	.05349	.06121	.05500	.22621
41	35-4-J	.07638	.07439	.07551	.06786	.29413
42	15-15-J	.02537	.02332	.02808	.02474	.10151
43	15-2-K	.06534	.05905	.05894	.05441	.23774
44	35-2-K	.08145	.07520	.07900	.08028	.31594

TABLE XV.

ENGINE PERFORMANCE—*Continued.*

Number	Laboratory Symbol.	Indicated Horse-power,					Steam Used per I.H.P. per Hour.		Dry Coal Used per I.H.P. per Hour.*
		Right Side.		Left Side.		Total.	Measured by Tank.	Measured by Indicator.	
		H.E.	C.E.	H.E.	C.E.		Lbs.	Lbs.	Lbs.
1	2	118	119	120	121	122	123	124	125
1	15–1$_b$–V	44.37	43.98	51.27	47.09	186.71	29.973	19.999	6.903
2	25–1–V	59.28	60.10	72.31	63.64	255.33	28.780	19.891	4.355
3	35–1–V	70.17	71.51	90.03	80.20	311.91	27.275	19.480	4.359
4	55–1–V	71.72	83.08	80.75	80.84	316.39	30.621	20.423	5.262
5	15–1–A	44.47	46.20	51.51	48.10	190.28	28.929	19.171	4.449
6	25–1–A	48.33	52.08	60.27	55.74	216.42	28.062	29.288	4.188
7	35–1–A	68.52	72.52	81.57	78.15	300.76	26.937	19.710	4.184
8	45-1-A	71.81	75.24	76.34	78.94	302.33	28.604	19.476	4.329
9	55–1–A	70.15	71.98	74.34	74.64	291.11	30.645	21.069	5.119
10	15-2-A	64.81	64.86	66.72	66.04	262.43	27.661	20.233	4.187
11	25-2-A	86.47	88.86	94.67	91.51	361.51	26.595	20.432	4.449
12	35–2–A	103.24	108.35	116.06	106.82	434.47	26.286	21.021	4.543
13	45–2–A	104.20	110.28	111.41	111.89	437.78	28.445	21.457	5.600
14	55–2–A	106.93	109.10	111.66	110.44	438.13	31.997	23.094	6.031
15	25–3–A	108.97	111.14	117.71	113.71	451.53	28.593	21.657	5.081
16	35–3–A	121.35	120.86	128.15	125.17	495.53	30.103	23.492	6.323
17	15–9–A	72.90	70.33	74.80	72.79	290.82	39.162	33.343	6.896
18	35–2–B	71.71	72.76	80.61	74.97	300.05	28.821	22.646	5.130
19	35–2–C	125.32	126.32	143.62	126.90	522.16	24.861	20.354	5.120
20	35–2–E	74.52	74.34	83.15	74.02	306.03	27.922	22.410	4.465
21	35–2–F	73.12	73.88	78.65	71.82	297.47	27.184	22.523	4.324
22	15–1–G	53.09	52.09	55.62	51.48	212.28	29.263	20.062	4.205
23	35–1–G	78.03	75.14	88.64	81.71	324.52	28.827	20.420	4.664
24	35–1$_b$–G	78.08	79.66	87.64	82.82	328.21	28.696	20.408	4.463
25	55–1–G	81.92	85.78	96.83	91.53	356.06	29.096	20.665	5.293
26	35–2–G	110.77	107.79	121.13	116.33	456.02	26.967	21.023	5.084
27	35–3–G	82.53	87.78	97.06	91.95	359.31	32.220	25.030	5.615
28	15–9–G	75.08	72.75	76.32	75.28	299.43	38.593	32.966	6.871
29	15–1–H	50.65	50.20	52.11	48.62	201.58	30.032	19.980	3.926
30	35–1–H	76.59	78.56	87.65	80.66	323.46	28.602	19.753	4.814
31	55–1–H	82.65	78.36	95.23	88.13	344.37	29.725	19.972	4.651
32	35–2–H	100.14	98.58	108.39	103.71	410.82	30.058	21.683	5.173
33	35–2$_b$–H	74.46	76.63	84.00	79.60	314.69	30.719	22.378	5.149
34	35–3–H	84.42	85.81	94.07	92.31	356.61	32.735	24.982	5.956
35	15–9–H	72.42	69.97	74.26	73.09	289.74	39.476	33.924	7.320
36	15–1–I	41.64	48.60	46.90	48.44	185.59	32.943	21.140	
37	35–1–I	68.39	85.04	80.55	87.17	321.15	29.366	21.317	
38	55–1–I	84.22	96.21	99.81	102.08	382.34	29.383	19.603	
39	15–9–I	65.42	65.98	67.32	66.57	265.30	40.865	35.393	
40	15–4–J	58.29	55.70	60.29	54.88	229.16	28.520	19.563	
41	35–4–J	90.86	89.02	105.06	95.82	380.76	26.772	19.458	
42	15–15–J	79.53	77.91	81.05	80.51	319.02	35.130	29.040	
43	15–2–K	0.01	54.81	59.71	56.94	231.47	28.810	20.310	
44	35–2–K	84.21	81.61	99.50	91.54	356.87	27.820	20.580	

* To obtain an approximate measure of the performance of the locomotive when using West Virginia or Pittsburgh coal, multiply column 125 by 0.8.

TABLE XVI.

ENGINE PERFORMANCE—(*Continued*).

RESULTS FROM INDICATOR-CARDS.

Number.	Laboratory Symbol.	Weight Steam per Revolution by Tank. Lbs.	Weight Mixture in Cylinder per Revolution. Lbs.	Per Cent of Mixture Present as Steam at Cut-off.	Per Cent of Mixture Present as Steam at Release.	Reëvaporation per Revolution. Lbs.	Condensation per Revolution. Lbs.	Reëvaporation per I.H.P. per Hour. Lbs.	Condensation per I.H.P. per Hour. Lbs.
1	**2**	**126**	**127**	**128**	**129**	**130**	**131**	**132**	**133**
1	15-1b-V	1.19011	1.47285	73.11	72.62		.00724		.18267
2	25-1-V	1.00752	1.31161	75.23	76.27	0.01367		0.39060	
3	35-1-V	0.76024	1.11706	79.73	80.55	.00919		0.32970	
4	55-1-V	0.53705	0.95820	80.18	81.34	.01110		0.63293	
5	15-1-A	1.13690	1.41663	72.18	72.22	.01057		0.39944	
6	25-1-A	0.78156	1.08338	76.37	77.73	.01478		0.53073	
7	35-1-A	0.70709	1.04562	80.12	81.86	.01817		0.70708	
8	45-1-A	0.57873	0.94651	79.94	81.18	.01185		0.58587	
9	55-1-A	0.48625	0.88476	80.15	82.82	.02367		1.49286	
10	15-2-A	1.56924	1.78283	76.97	76.36		.01081		.19032
11	25-2-A	1.25553	1.50064	80.30	80.69	.00453		0.09615	
12	35-2-A	1.00165	1.30263	83.41	84.60	.01549		0.40628	
13	45-2-A	0.84323	1.18184	81.24	82.48	.01459		0.49207	
14	55-2-A	0.76357	1.14063	79.44	81.37	.02204		0.92330	
15	25-3-A	1.66629	1.88635	79.46	78.58		.01668		.26461
16	35-3-A	1.35278	1.66256	81.64	82.13	.00810		0.18023	
17	15-9-A	2.39542	2.48467	86.61	85.67		.02329		.38076
18	35-2-B	0.76518	1.05159	80.32	84.40	.04284		1.61392	
19	35-2-C	1.14142	1.45159	84.28	85.74	.02120		0.46180	
20	35-2-E	0.73353	1.02511	81.55	85.87	.04439		1.68960	
21	35-2-F	0.72044	1.00404	81.88	87.70	.05839		2.20343	
22	15-1-G	1.27736	1.52407	72.38	73.65	.01923		0.44054	
23	35-1-G	0.81905	1.13317	77.68	78.92	.01403		0.49379	
24	35-1b-G	0.82510	1.14282	77.25	79.15	.02165		0.75295	
25	55-1-G	0.57080	0.95449	82.02	82.57	.00621		0.31655	
26	35-2-G	1.09486	1.37754	81.67	82.48	.01121		0.27611	
27	35-3-G	1.01506	1.28049	81.43	82.32	.01134		0.35995	
28	15-9-G	2.44208	2.52864	87.23	85.92		.03327		.52578
29	15-1-H	1.23895	1.50283	70.86	72.41	.02334		0.56587	
30	35-1-H	0.81161	1.10193	75.39	77.21	.02003		0.70587	
31	55-1-H	0.57981	0.93540	77.99	79.66	.01563		0.80128	
32	35-2-H	1.06528	1.32022	75.87	77.52	.02177		0.61425	
33	35-2b-H	0.84097	1.08826	77.18	79.02	.02001		0.73092	
34	35-3-H	1.00251	1.21165	79.53	80.41	.01056		0.34481	
35	15-9-H	2.37938	2.46511	87.66	86.43		.03052		.50635
36	15-1-I	1.25908	1.04869	82.81	83.80	.01243		0.32532	
37	35-1-I	0.82364	1.10067	74.71	76.46	.01942		0.69249	
38	55-1-I	0.64438	0.93939	75.64	77.16	.01427		0.65088	
39	15-9-I	2.27697	2.37070	88.26	87.19		.02516		.45132
40	15-4-J	1.31060	1.53681	71.58	73.22	.02516		0.54740	
41	35-4-J	0.89140	1.18553	76.07	79.46	.04015		1.20500	
42	15-15-J	2.37320	2.47470	85.28	83.38		.04696		.69510
43	15-2-K	1.35150	1.58920	73.47	70.49	.02316		0.49379	
44	35-2-K	0.91551	1.23150	78.17	78.18	.00677		0.02132	

TABLE XVII.
LOCOMOTIVE PERFORMANCE.

Number.	Laboratory Symbol.	Draw-bar Pull Equivalent to Average M.E.P. Lbs.	Machine Friction of Engine in Terms of Average M.E.P.	Machine Friction of Engine in Terms of Draw-bar Pull Lbs.	Machine Friction of Engine in Terms of I.H.P.	Draw-bar or Dynamometer H.P.	Mechanical Efficiency of Engine. Per Cent.	I.H.P. per Square Foot of Heating Surface.	I.H.P. per Square Foot of Grate Surface.
1	2	134	135	136	137	138	139	140	141
1	15-1b-V	4,806	5.29	579	22.5	164.2	87.9	.154	10.8
2	25-1-V	4,236	5.29	579	34.9	220.4	86.2	.210	14.8
3	35-1-V	3,375	5.29	579	53.5	258.4	82.8	.256	17.1
4	55-1-V	2,123	5.29	579	86.3	230.1	72.7	.260	18.3
5	15-1-A	4,759	5.29	579	23.1	167.2	87.9	.157	11.0
6	25-1-A	3,372	5.29	579	37.1	179.3	82.9	.178	12.5
7	35-1-A	3,179	5.29	579	54.8	246.0	81.8	.248	17.4
8	45-1-A	2,449	5.29	579	71.4	230.9	76.4	.249	17.5
9	55-1-A	1,921	5.29	579	87.7	203.4	69.9	.239	16.9
10	15-2-A	6,867	4.18	458	17.5	244.9	93.2	.226	15.2
11	25-2-A	5,717	4.18	458	28.9	332.6	92.0	.297	20.9
12	35-2-A	4,615	4.18	458	43.1	391.4	90.1	.357	25.2
13	45-2-A	3,590	4.18	458	55.8	382.0	87.3	.359	25.4
14	55-2-A	2,889	4.18	458	69.5	368.6	84.1	.361	25.3
15	25-3-A	7,057	3.57	391	25.0	426.5	94.5	.370	26.1
16	35-3-A	5,441	3.57	391	35.6	459.9	92.8	.407	28.7
17	15-9-A	7,409	1.69	185	7.3	283.5	97.5	.239	16.8
18	35-2-B	3,214	4.18	458	42.8	257.3	85.7	.247	17.4
19	35-2-C	5,558	4.18	458	43.0	479.2	91.8	.430	30.2
20	35-2-E	3,180	4.18	458	44.1	261.9	85.6	.252	17.7
21	35-2-F	3,209	4.18	458	42.4	255.1	85.7	.244	17.2
22	15-1-G	5,283	5.29	579	23.2	189.1	89.1	.174	12.3
23	35-1-G	3,439	5.29	579	54.6	269.9	83.2	.267	18.8
24	35-1b-G	3,482	5.29	579	54.6	273.6	83.4	.270	18.9
25	55-1-G	2,375	5.29	579	86.8	269.3	75.6	.293	20.6
26	35-2-G	4,915	4.18	458	42.5	413.5	90.7	.375	26.4
27	35-3-G	3,814	3.57	391	36.8	322.5	89.8	.296	20.8
28	15-9-G	7,658	1.69	185	7.2	292.2	97.6	.246	17.4
29	15-1-H	4,997	5.29	579	23.4	178.2	88.4	.166	11.7
30	35-1-H	3,436	5.29	579	54.5	269.0	83.1	.266	18.6
31	55-1-H	2,362	5.29	579	84.4	260.0	75.5	.284	20.0
32	35-2-H	4,291	4.18	458	43.8	367.0	89.3	.338	23.8
33	35-2b-H	3,315	4.18	458	43.5	271.2	86.2	.259	18.2
34	35-3-H	3,707	3.57	391	37.6	319.0	89.5	.294	20.7
35	15-9-H	7,299	1.69	185	7.3	282.4	97.5	.238	16.8
36	15-1-I	4,632	5.29	579	23.2	162.4	87.3	.153	10.8
37	35-1-I	3,401	5.90	646	61.0	260.2	81.1	.264	18.6
38	55-1-I	2,591	5.90	646	92.9	289.4	75.6	.315	22.2
39	15-9-I	6,752	1.69	185	7.3	258.0	97.3	.219	15.4
40	15-4-J	5,562	5.29	579	23.8	205.4	89.7	.189	13.3
41	35-4-J	4,031	5.50	602	56.8	324.0	85.1	.313	22.1
42	15-15-J	8,185	2.00	219	8.8	310.2	97.2	.263	18.5
43	15-2-K	5,678	4.90	536	21.8	209.7	90.6	.191	13.4
44	35-2-K	3,995	4.90	536	48.0	308.9	86.6	.294	20 7

TABLE XVIII.

LOCOMOTIVE PERFORMANCE—(*Continued*).

Number.	Laboratory Symbol.	Steam Used per D.H.P. per Hour. Lbs.	Dry Coal Used per D.H.P. per Hour.* Lbs.	B.T.U. Used by Engine. Per Minute.	B.T.U. Used by Engine. Per I.H.P. per Minute.	B.T.U. Used by Engine. Per D.H.P. per Minute.	Coal Used per Mile Run. Lbs.	D.H.P. Developed per Square Foot. Heating Surface.	D.H.P. Developed per Square Foot. Grate Surface.
1	2	142	143	144	145	146	147	148	149
1	15-1b-V	34.08	4.86	107,890	5,778	6,570	55.88	.135	9.5
2	25-1-V	33.34	5.06	141,610	5,446	6,425	49.36	.181	12.9
3	35-1-V	32.92	5.24	164,275	5,267	6,357	39.32	.212	14.9
4	55-1-V	42.11	6.95	185,707	5,869	8,071	29.80	.189	13.4
5	15-1-A	32.92	4.69	106,250	5,584	6,354	56.66	.138	9.7
6	25-1-A	33.87	4.90	116,805	5,397	6,514	37.78	.148	10.4
7	35-1-A	32.93	5.14	156,083	5,189	6,344	35.55	.203	14.3
8	45-1-A	37.45	5.95	166,053	5,492	7,191	28.28	.190	13.4
9	55-1-A	43.86	7.01	170,938	5,872	8,404	26.22	.167	11.8
10	15-2-A	29.64	4.48	139,656	5,322	5,702	76.69	.202	14.2
11	25-2-A	28.91	4.77	184,636	5,107	5,551	67.82	.274	19.3
12	35-2-A	29.18	5.19	219,844	5,060	5,616	56.00	.322	22.7
13	45-2-A	32.59	6.06	238,677	5,452	6,248	53.60	.315	22.1
14	55-2-A	38.03	7.59	267,981	6,102	7,270	46.47	.304	21.3
15	25-3-A	30.27	5.77	248,051	5,493	5,815	95.60	.351	24.7
16	35-3-A	32.44	6.79	285,931	5,770	6,217	91.74	.379	26.7
17	15-9-A	40.17	7.13	218,483	7,512	7,706	136.19	.233	16.4
18	35-2-B	33.62	5.36	166,387	5,545	6,466	44.05	.213	14.9
19	35-2-C	27.09	5.23	251,132	4,809	5,228	76.02	.395	27.8
20	35-2-E	32.63	5.22	164,943	5,389	6,298	37.94	.216	15.3
21	35-2-F	31.71	4.97	156,202	5,251	6,125	37.00	.210	14.8
22	15-1-G	32.85	4.77	119,319	5,620	6,310	59.26	.156	11.0
23	35-1-G	34.66	5.65	179,366	5,527	6,645	42.78	.222	15.6
24	35-1b-G	34.42	5.63	180,721	5,506	6,605	41.43	.225	15.9
25	55-1-G	38.38	6.53	198,585	5,577	7,374	33.52	.222	15.6
26	35-2-G	29.74	5.48	235,413	5,162	5,680	66.64	.341	24.0
27	35-3-G	35.89	6.40	221,649	6,168	6,873	57.12	.266	18.7
28	15-9-G	39.55	7.07	221,657	7,403	6,169	140.38	.241	16.9
29	15-1-H	33.97	4.87	115,673	5,738	6,491	52.29	.147	10.3
30	35-1-H	34.39	5.57	177,190	5,478	6,587	44.10	.222	15.6
31	55-1-H	39.37	5.89	196,619	5,709	7,562	29.30	.214	15.1
32	35-2-H	33.90	6.20	236,330	5,752	6,439	59.19	.302	21.3
33	35-2b-H	35.64	5.91	185,912	5,907	6,855	45.51	.223	15.7
34	35-3-H	36.60	6.57	223,693	6,272	7,012	58.89	.261	18.5
35	15-9-H	40.50	7.19	219,289	7,568	7,783	142.46	.233	16.4
36	15-1-I								
37	35-1-I								
38	55-1-I								
39	15-9-I								
40	15-4-J								
41	35-4-J								
42	15-15-J								
43	15-2-K								
44	35-2-K								

* To obtain an approximate measure of the performance of the locomotive, when using West Virginia or Pittsburgh coal, multiply the values of column 143 by 0.8.

TABLE XIX.

LOCOMOTIVE PERFORMANCE—(*Continued*).

The values of this table were computed on the assumption of uniform fire conditions. The process involved is developed in Chapter VI.

Number.	Laboratory Symbol.	Equivalent Evaporation from and at 212° F. per Hour. Lbs.	Coal Fired per Hour. Lbs.	Coal Burned per Square Foot Grate Surface per Hour. Lbs.	Water (by Tank) Evaporated per Pound of Coal. Lbs.	Equivalent Evaporation from and at 212° per Pound of Coal. Lbs.	Coal per I.H.P. per Hour. Lbs.	Coal per D.H.P. per Hour. Lbs.	B.T.U. Taken up by Boiler per Pound of Coal.
1	**2**	**150**	**151**	**152**	**153**	**154**	**155**	**156**	**157**
1	15-1b-V	6,762	798.0	46.2	7.04	8.44	4.270	4.86	8,151.3
2	25-1-V	8,864	1,113.1	64.5	6.62	7.93	4.359	5.05	7,658.8
3	35-1-V	10,286	1,349.3	78.9	6.32	7.59	4.325	5.22	7,330.4
4	55-1-V	11,650	1,590.3	92.1	6.10	7.27	5.026	6.91	7,021.5
5	15-1-A	6,660	791.6	45.8	6.98	8.37	4.160	4.73	8,083.7
6	25-1-A	7,318	876.6	50.8	6.95	8.31	4.050	4.89	8,025.8
7	35-1-A	9,751	1,261.2	73.0	6.44	7.71	4.193	5.13	7,446.3
8	45-1-A	10,395	1,367.3	79.2	6.34	7.56	4.522	5.92	7,300.1
9	55-1-A	10,690	1,416.8	82.1	6.31	7.49	4.867	6.97	7,253.8
10	15-2-A	8,745	1,093.1	63.3	6.66	7.96	4.165	4.46	7,687.7
11	25-2-A	11,555	1,590.0	92.1	6.06	7.23	4.398	4.78	6,982.7
12	35-2-A	13,733	2,028.6	117.7	5.64	6.75	4.669	5.18	6,519.1
13	45-2-A	14,926	2,301.0	133.4	5.42	6.46	5.256	6.02	6,239.1
14	55-2-A	16,773	2,802.3	162.2	5.01	6.02	6.396	7.78	5,814.1
15	25-3-A	15,515	2,444.4	141.6	5.29	6.02	5.414	5.73	6,103.8
16	35-3-A	17,878	3,099.0	179.6	4.82	5.75	6.254	6.74	5,553.3
17	15-9-A	13,670	2,040.9	118.2	5.59	6.67	7.018	7.20	6,441.8
18	35-2-B	10,404	1,371.5	79.5	6.32	7.56	4.571	5.33	7,300.4
19	35-2-C	15,709	2,496.0	144.6	5.21	6.27	4.780	5.21	6,055.5
20	35-2-E	10,324	1,355.3	78.5	6.32	7.58	4.428	5.17	7,320.7
21	35-2-F	9,785	1,263.4	73.2	6.42	7.71	4.247	4.95	7,446.3
22	15-1-G	7,476	900.0	52.1	6.93	8.27	4.239	4.76	7,987.1
23	35-1-G	11,226	1,517.0	87.9	6.18	7.36	4.674	5.62	7,108.3
24	35-1b-G	11,312	1,532.2	88.8	6.16	7.34	4.668	5.60	7,089.0
25	55-1-G	12,425	1,747.5	101.2	5.94	7.07	4.908	6.49	6,828.2
26	35-2-G	14,725	2,256.0	130.7	5.46	6.50	4.947	5.46	6,277.7
27	35-3-G	13,866	2,052.6	113.1	5.65	6.72	5.712	6.36	6,490.2
28	15-9-G	13,869	2,049.0	118.8	5.65	6.72	6.843	7.01	6,490.2
29	15-1-H	7,248	866.5	50.2	7.01	8.32	4.298	4.86	8,035.2
30	35-1-H	11,090	1,490.5	86.3	6.22	7.40	4.608	5.54	7,146.9
31	55-1-H	12,302	1,675.8	97.1	6.12	7.34	4.866	6.45	7,052.2
32	35-2-H	14,769	1,959.8	113.6	6.31	7.50	4.770	5.34	7,243.5
33	35-2b-H	11,633	1,593.0	92.3	6.08	7.26	5.062	5.87	7,011.7
34	35-3-H	13,993	2,080.6	120.7	5.62	6.69	5.834	6.52	6,461.2
35	15-9-H	13,752	2,031.5	117.8	5.64	6.72	7.012	7.19	6,490.2
36	15-1-I								
37	35-1-I								
38	55-1-I								
39	15-9-I								
40	15-4-J								
41	35-4-J								
42	15-15-J								
43	15-2-K								
44	35-2-K								

II. LOCOMOTIVE PERFORMANCE, A TYPICAL EXHIBIT.

CHAPTER V.

LOCOMOTIVE PERFORMANCE AS AFFECTED BY CHANGES IN SPEED AND CUT-OFF.

29. **Purpose.**—It is the purpose of the present chapter to define and summarize certain fundamental facts concerning cylinder performance. It shows, by means of data derived from experiments, the effect of changes in speed and cut-off upon the power and efficiency of a locomotive when running under a wide-open throttle.

30. **The Tests** which form the basis for this discussion, twelve in number, are those which, in the tabulated record of Chapter IV., are designated as Series A. They cover practically all conditions of speed and cut-off at which it is possible to operate continuously the experimental locomotive with a wide-open throttle. Five tests were run with the reverse lever in the first notch, giving a cut-off of approximately 25 per cent, the speeds being 15, 25, 35, 45 and 55 miles per hour, respectively; five were run with the reverse lever in the second notch, giving a cut-off of approximately 35 per cent, the speeds being the same as for the shorter cut-off, and two were run with the reverse-lever in the third notch, giving a cut-off of approximately 45 per cent., the speeds being 25 and 35 miles per hour, respectively. The steam pressure, the setting of the valves, the fuel used, and the fireman were the same for all tests. Excepting as to speed and cut-off, which were as specified above, the conditions affecting all tests were as nearly as possible identical.

Table XX. presents the laboratory symbol of the several tests and a summary of the more important controlling conditions.

TABLE XX.

ESSENTIAL CONDITIONS.

Designation of Test.	Speed.		Cut-off.		Steam Pressure.		Throttle.	Fuel.	Fireman.
	Miles per Hour.	Revolutions per Minute.	Position of Reverse-lever, Notches from Center Forward.	Approximate Per Cent of Stroke.	In Boiler, Pounds.	In Dry Pipe, Pounds.			
15-1-A	15.00	80.7	1	25	125.9	124.6	Wide open	Brazil Indiana Block	Chas. Reyer
25-1-A	24.07	129.5	1	25	120.0	113.4	"	"	"
35-1-A	35.49	191.0	1	25	129.7	127.2	"	"	"
45-1-A	46.29	249.1	1	25	128.8	124.9	"	"	"
55-1-A	56.83	305.7	1	25	124.9	121.3	"	"	"
15-2-A	14.33	77.1	2	35	129.5	125.1	Wide open	Brazil Indiana Block	Chas. Reyer
25-2-A	23.72	127.6	2	35	129.3	124.9	"	"	"
35-2-A	35.31	190.0	2	35	131.6	120.7	"	"	"
45-2-A	45.74	246.1	2	35	126.7	121.1	"	"	"
55-2-A	56.81	306.0	2	35	124.0	118.8	"	"	"
25-3-A	24.00	129.1	3	45	127.2	123.2	Wide open	Brazil Indiana Block	Chas. Reyer
35-3-A	34.15	183.8	3	45	125.3	122.1	"	"	"

31. **The Valves and their Setting.**—The slide-valves used in this series of tests had $\frac{3}{4}''$ outside lap, $\frac{1}{32}''$ inside lap, and a maximum travel of 5.57″. Their setting was that which, after considerable experimenting, had been found to produce the most efficient results. When the cut-off was 25 per cent of the stroke, they gave $\frac{1}{32}''$ lead. At full stroke, the lead was nil. (See A setting, Chapter III.)

32. **Indicator-cards.**—In Fig. 61 are presented average cards from the left side of the locomotive for each test. These cards, when studied in connection with the results derived from them, will serve to disclose the effect upon the distribution of steam in the cylinder, of changes in speed and cut-off. Comparisons along vertical lines show the effect of changes in speed at constant cut-off, along horizontal lines, of changes in cut-off at constant speed.

Examining the cards along either of the vertical lines, as, for example, along that which represents a cut-off of 25 per cent, it appears

that as the speed is increased, the size of the card is diminished. At very high speeds the card is comparatively small. This change occurs notwithstanding the fact that there is no change in cut-off or in any other event of the stroke. It is due wholly to wiredrawing past the valves. While the extent of port opening may be unaffected by changes in speed, the period of opening is reduced as the speed is

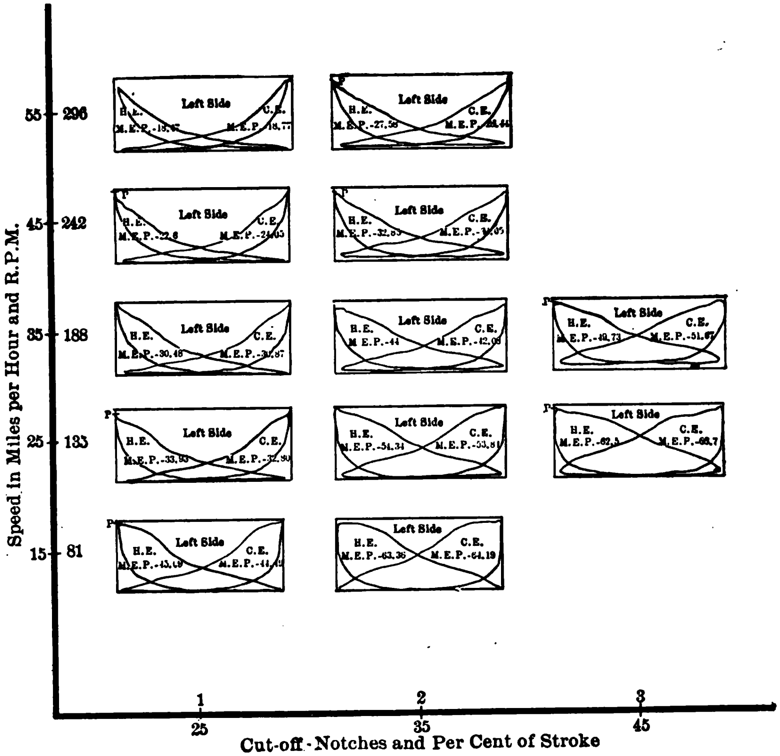


FIG. 61.

increased, from which it necessarily follows that there is admitted less steam per stroke as the speed is increased. A further discussion of this matter will be found in Chapter XVII., which deals especially with valve-gears. It will be sufficient for the present to observe that the size of the card is controlled to a limited extent only by the cut-off. Thus, by reference to Fig. 61, it will be found easy to select cards representing a cut-off of 35 per cent which are actually smaller than others representing a cut-off of 25 per cent.

An interesting illustration of the influence of speed upon the size of the card is afforded when an attempt is made to operate a locomotive with the reverse-lever in its extreme forward position at high speed. Fig. 62 represents two cards, both taken with the reverse-lever in its extreme forward position. The dotted line is the normal card taken at low speed, the full line a card taken at 35 miles an hour. It will be seen that the total pressure range for the card at speed is divided into three parts, more than a third representing the loss in passing

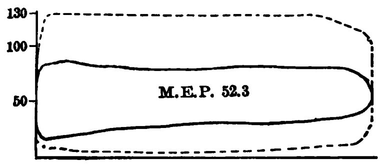


FIG. 62.

from boiler to cylinder, a quarter from cylinder to exhaust, and approximately a third only the mean effective pressure. It is evident that speed as well as cut-off have an important influence on mean effective pressure.

In this connection, also, attention should be called to the changed form of the card which results from change of speed (Fig. 61). At low speeds the events of the stroke are rather clearly marked, but as the speed is increased these become less and less distinct. These statements will be sufficient to show how much more complicated is the study of locomotive performance than is that of stationary engines designed to run at fixed speeds, for in locomotive service the element of speed appears as an ever-present influence affecting other factors; for example, the mean effective pressure, the distribution of steam, the thermodynamic efficiency, and even the machine friction.

33. Events of the Stroke.—In the early work upon the locomotive it was assumed that a fixed location of the reverse-lever would serve to maintain all events of the stroke constant. It was soon found, however, that the cut-off, for example, could not be depended upon to remain constant even though the reverse-lever was not moved, but that a change in the speed of the locomotive or in the lubrication of its valves was quite sufficient to materially affect the time of action of the valves. This is to be accounted for by the fact that the width of port-opening is small, the force required

to move the valve is considerable, and the mechanis mwhich drives it is not sufficiently stiff nor so free from lost motion as to impart to the valve a motion which is absolutely positive. The precise values representing the several events of the stroke for each of the twelve tests under consideration, as obtained from indicator-cards taken under the conditions defined, are given in Table XXI. Cards taken at high speeds do not show clearly the point of admission, the value of which is therefore omitted from the table.

TABLE XXI.

EVENTS OF STROKE AS DETERMINED FROM INDICATOR-CARDS.

The values given are the averages for the four cylinder-ends.

Designation of Test.	Per Cent of Stroke.			
	Admission.	Cut-off.	Release.	Beginning of Compression.
15-1-A	3.25	24.68	71.08	33.37
25-1-A	3.25	24.68	71.08	33.37
35-1-A	3.25	24.68	71.08	33.37
45-1-A	2.15	22.43	73.34	22.69
55-1-A	——	22.61	74.20	22.62
15-2-A	1.50	37.80	75.80	21.71
25-2-A	1.53	34.37	74.51	27.49
35-2-A	1.70	33.69	77.94	28.62
45-2-A	1.75	33.07	78.25	24.50
55-2-A	——	35.40	76.90	27.32
25-3-A	1.32	44.22	79.60	16.62
35-3-A	1.51	43.82	80.54	24.49

34. **Wiredrawing** between boiler and cylinder in a locomotive is unavoidable, the extent of its influence in any given locomotive depending upon the rate at which steam is used. The areas of the wide-open throttle, steam-pipe, branch pipes, etc., for locomotive Schenectady No. 1 are shown graphically by Fig. 54, Chapter III. The facts concerning the drop in the pressure of the steam from the boiler to the exhaust-pipe, with related data of interest for the twelve tests under consideration, are shown by Table XXII.

35. **Mean Effective Pressure,** as affected by speed and cut-off, is shown diagrammatically by Fig. 63. If the values of this figure are compared along vertical lines, the very striking effect resulting from changes in speed may be seen. With a given cut-off each

increment in speed reduces the amount of work done per stroke, and while values measuring such changes depend somewhat upon the size of ports and the characteristics of the valve-gear employed, the presence of such a change is unavoidable in the action of a locomotive. (Chapter XVII.)

TABLE XXII.

DROP IN STEAM PRESSURE.

WIDE-OPEN THROTTLE. EXHAUST TIP DOUBLE 3″ DIAM.

Designation of Test.	Steam Exhausted per Hour, Lbs.	Maximum Port-opening, Inches.		Pressure of Steam in					
						Cylinders			
		Steam.	Exhaust.	Boiler.	Branch Pipe.	At Cut-off.	At Release.	At Compression.	Least Back.
15-1-A	5,505	.20	.92	125.9	124.6	91.0	28.4	6.4	1.3
25-1-A	6,073	.20	.92	120.0	113.4	69.7	19.9	8.1	1.8
35-1-A	8,102	.20	.92	129.7	127.2	70.6	20.4	10.9	3.0
45-1-A	8,648	.20	.92	128.8	124.9	67.9	16.1	23.4	2.7
55-1 A	8,921	.20	.92	124.9	121.3	59.8	14.4	26.9	2.8
15-2-A	7,259	.25	.97	129.5	125.1	87.9	39.7	7.5	0.8
25-2-A	9,614	.25	.97	129.3	124.9	82.1	34.2	6.8	2.8
35-2-A	11,420	.25	.97	131.6	120.7	73.4	27.6	11.0	5.5
45-2-A	12,451	.25	.97	126.7	122.1	63.9	22.5	18.3	5.5
55-2-A	14,019	.25	.97	124.0	181.8	55.0	21.1	19.1	5.8
25-3-A	12,910	.30	1.02	127.2	123.2	83.8	42.1	12.7	3.5
35-3-A	14,917	.30	1.02	125.3	122.1	74.7	37.0	15.3	9.1

Another fact of interest in connection with Fig. 63 is that the values given cover practically the entire range of action under which the experimental locomotive can be operated with a wide-open throttle. An attempt to run a test at 15 miles and 10″ cut-off resulted in so high a tractive power that the drivers slipped, while a test at 45 miles and 10″ cut-off could not be run because of the failure of the boiler to supply steam.

36. The Indicated Horse-power for different speeds and cut-offs is shown by Fig. 64. Here it will be seen that at constant cut-off, the power increases with increase of speed up to a certain point, after which it remains practically constant. In the case of the locomotive experimented upon, the maximum power was practically reached at a speed of 35 miles an hour. The limit, however, as applied to

locomotives in general will depend upon the proportion of the cylinders, the diameter of the drivers, and the capacity of the boiler, and while this point may not always be as clearly defined as it is by the data from Schenectady No. 1, its presence will appear in all locomotives which are designed to run at high speeds of rotation.

It will be seen that the highest power reached was 496 horse. Prior to the tests herein described, no one knew what was the maximum power of a locomotive under constant operating con-

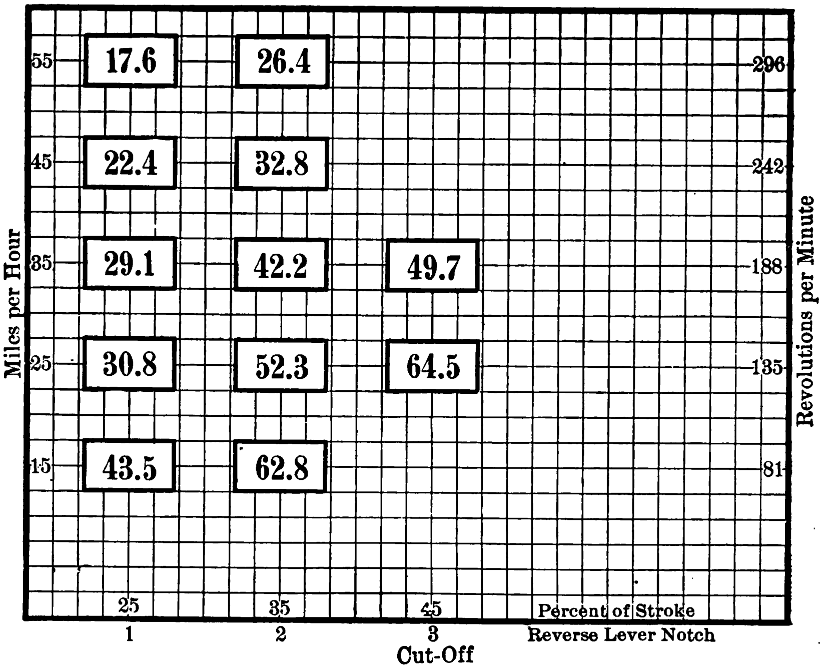


FIG. 63.—Mean Effective Pressure.

ditions, though it was not uncommon for locomotives of the size of Schenectady No. 1 to be credited with as high as 800 horse-power. The facts presented by Fig. 64 and the fuller exhibit of Table XV. suggest that such an estimate is too high, though if a superior grade of coal had been used instead of the light and friable Brazil block, it is probable that the maximum could have been made to approach 600. It may easily be shown, by an analysis based on the dimensions of the locomotive and the experimental facts herein presented, that for all speeds below 18 miles an hour the cut-off,

with a fully open throttle, is limited by the adhesion of the drivers; for speeds above 18 miles the limit is found in the capacity of the boiler to supply steam. The cut-off which at any given speed will serve under a fully open throttle to operate the experimental locomotive at 500 horse-power is that shown by the curve, Fig. 64.

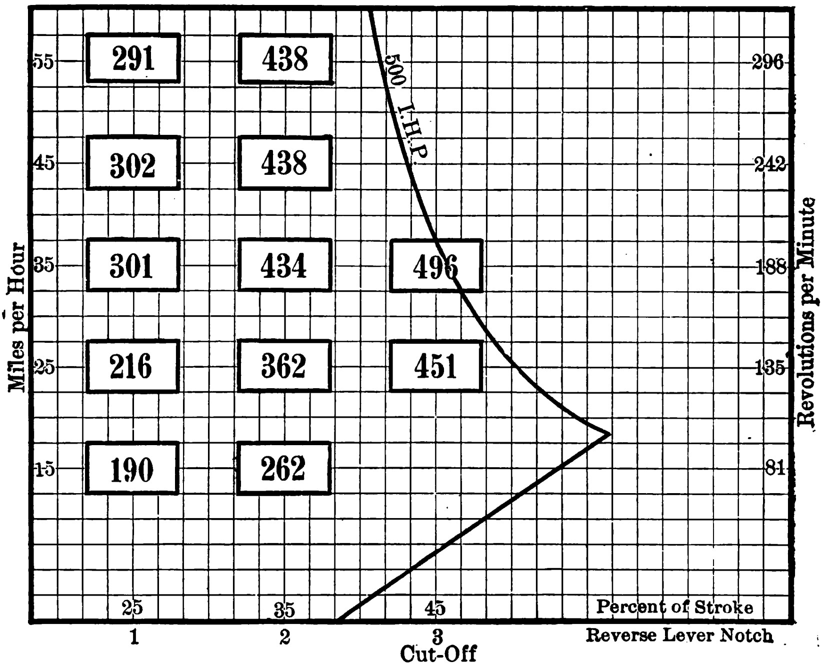


FIG. 64.—Indicated Horse-power.

It is well, also, to review the values of Fig. 64 with reference to a matter which is always of interest to locomotive designers, namely, that of controlling speed by the reverse-lever *vs.* the throttle. The reverse-lever quadrant of Schenectady No. 1 was notched at intervals of ¾ of an inch, and the first three notches forward from the center gave 25, 35, and 45 per cent cut-off respectively. Fig. 64 discloses the fact that at all speeds a change of one notch in the reverse-lever position makes a very marked alteration in the power output, the differences ranging from 70 to 146 horse-power. Between the first and second notches there is an increase of about 50 per cent over the power developed with the lever in the first notch. Evidently no very fine gradation of power, and consequently of speed, can be secured by

manipulating the reverse-lever of this locomotive. Quadrants of the modern locomotive are, however, much more closely notched, and as, in present-day practice, locomotives are allowed to go as fast as they will whatever load may be attached to them, the question outlined now has much less significance than formerly.

37. **The Steam Consumption** per horse-power hour for different speeds and cut-offs is shown numerically by Fig. 65 and graphically

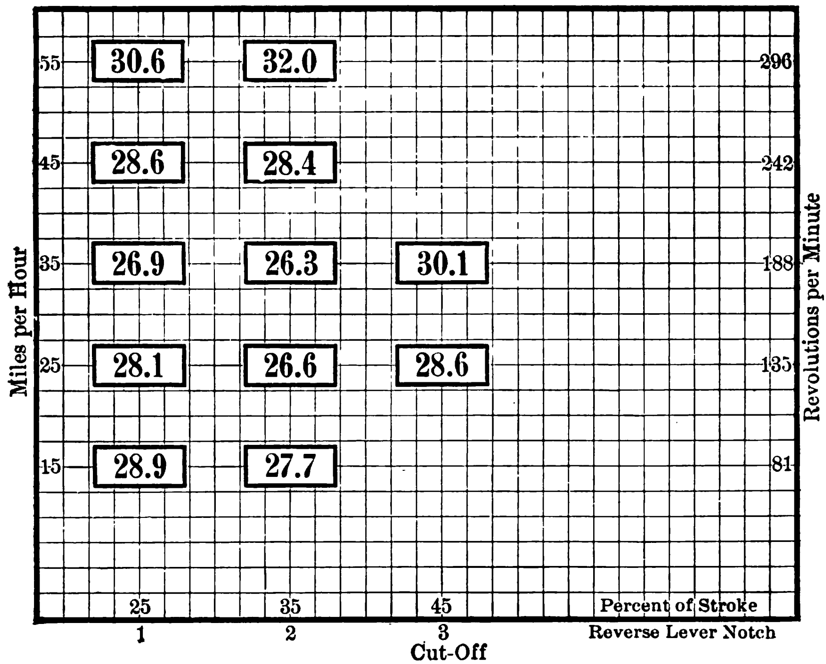


FIG. 65.—Steam per I.H.P. per Hour.

by Fig. 66. Engineers unfamiliar with the performance of the locomotive have often characterized it as an extremely wasteful engine, whereas from Fig. 65 it appears that its performance compares favorably with that of any other class of single-cylinder, non-condensing engine. With open throttle, the consumption of steam per indicated horse-power does not under any conditions of speed or cut-off exceed 32 pounds, and under favorable conditions it falls to about 26 pounds. In this connection it should be noted that Schenectady No. 1 carried but 140 pounds pressure, and that the tests were run at about 130. When favored by a higher pressure, this engine has given one horse-

power on a consumption of less than 25 pounds of steam per hour.

The results show that the minimum consumption is obtained at the 35 per cent cut-off, or, say, one-third stroke; this for an engine carrying 140 pounds of steam. The losses resulting from the employment of a shorter cut-off are, however, slight as compared with those which attend a lengthening of the cut-off. An exception to this statement is, however, to be found at high speed, where the 25 per cent cut-off is

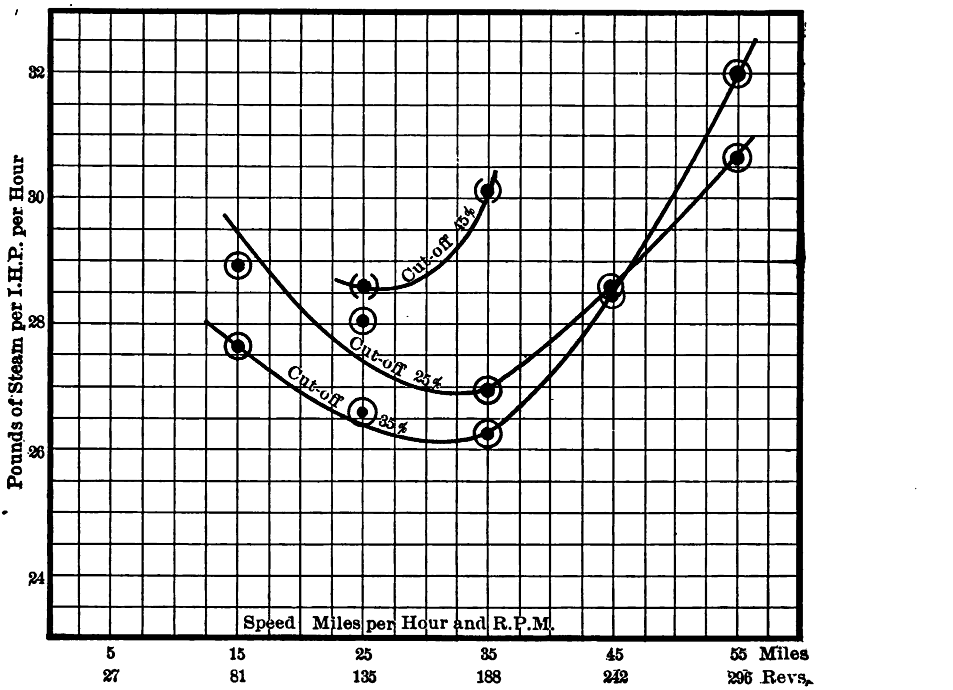


FIG. 66.—Steam per I.H.P. per Hour.

better than the 35 per cent. It should be said, also, that experiments involving higher steam pressures go to show that the most economical cut-off for all speeds is somewhere between one-quarter and one-third stroke. It appears, also, whatever may be the cut-off, that, beginning with slow motion, a gradual increase of speed is attended by a decrease in the amount of steam required up to a certain point, after which the consumption increases. Both an inspection of the values of Fig. 65 and a glance at the curves of Fig. 66 show the most economi-

cal performance to have been obtained at a speed of but 35 miles per hour, or 188 revolutions per minute. In explanation of this fact, it would appear that below this limit any increase of speed is of advantage through reduced cylinder condensation, an advantage which is often assumed to attend all increase of piston speed, but above this limit some other influence enters which is so strong in its effect as to more than neutralize the advantage of the higher piston speed. This neutralizing influence is without doubt the wiredrawing, the presence of which becomes more and more marked as the speed rises.

38. **Critical Speed.**—In the two paragraphs immediately preceding, attention has been called to the fact that the maximum power and the maximum efficiency of locomotive Schenectady No. 1 were found at a speed of approximately 35 miles an hour, or 190 revolutions per minute. To give significance to the conditions involved, the author has called that speed at which the power and efficiency of a locomotive become maximum the "critical speed." It will, of course, not always be found at 35 miles an hour, for much depends upon the diameter of drivers, but it will be found not far from 200 revolutions per minute. With higher steam pressures the limit is probably raised somewhat, and with a superior valve-gear the point may not be quite so well marked as in the data, but the preceding statement will be found substantially true in its application to all simple locomotives.

39. **Cylinder Condensation.**—The effect of changes in speed and cut-off upon the percentage of the total steam used, which is shown by the indicator, is given by Fig. 67. Two facts are clearly presented by this figure. The first is that the percentage of the steam used which is accounted for by the indicator, is greatest at the critical speed. This appears to be true for all cut-offs. The data well illustrate a rather commonly accepted theory, that for a given cut-off anything which tends to suppress cylinder condensation improves the performance of the engine. Fig. 67 shows that either increasing the speed above or diminishing it below 35 miles an hour increases the condensation, and it has already been shown (Fig. 65) that similar changes operate to increase the water consumption of the engine.

A second fact, which is apparent from Fig. 67, is that at constant speed any increase of cut-off increases the percentage of steam shown by the indicator, or diminishes the condensation. The data show some exceptions to this rule, but they involve values which are small. The tendency of the exhibit is clear. It will be elsewhere shown (throttling test, Chapter XX.) that by throttling and making the

cut-off very late in the stroke more than 90 per cent of all the steam used is shown by the indicator.

Dealing with the more detailed action which marks the interchange of heat between the steam and the walls of the cylinder, attention is first called to the percentage of mixture present as steam at cut-off. (Fig. 68.) The difference between the values given and 100 will represent the percentage of the mixture which is water. Here

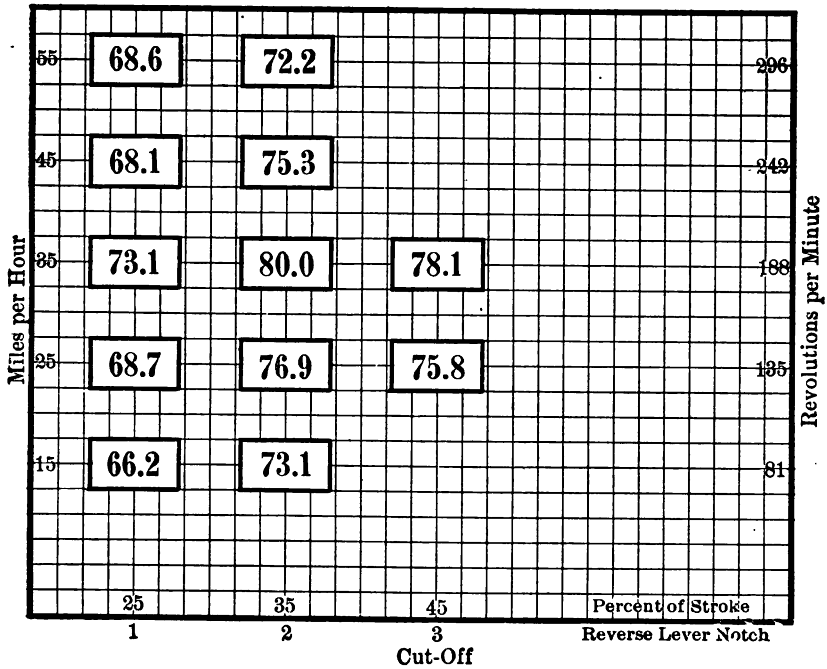


FIG. 67.—Percentage of Total Steam shown by Indicator.

again, it appears that at any given cut-off the percentage of the mixture which is present as steam is nowhere greater than at the critical speed. Other values of interest are shown by Table XXIII. Thus, the last four columns of this table show the extent of the reëvaporation during expansion when the cut-off is constant. This reëvaporation in all cases increases with increase of speed, while at constant speed lengthening the cut-off diminishes the reëvaporation. For example, referring to the 25-mile series, the test with the reverse-lever in the first notch gave a reëvaporation per revolution of .0148; with the reverse-lever

in the second notch the reëvaporation decreased to .0045 pound; and with the reverse-lever in the third notch all reëvaporation disappeared, and the condensation amounted to .0167 pound.

The weight of mixture in the cylinder per revolution and the weight exhausted (Table XXIII.) shows, when compared, the character of the exhaust action at different speeds. For example, referring to the series for which the reverse-lever was in the first notch, at 15 miles an hour, of the 1.4 pounds in the cylinder, 1.1 pounds

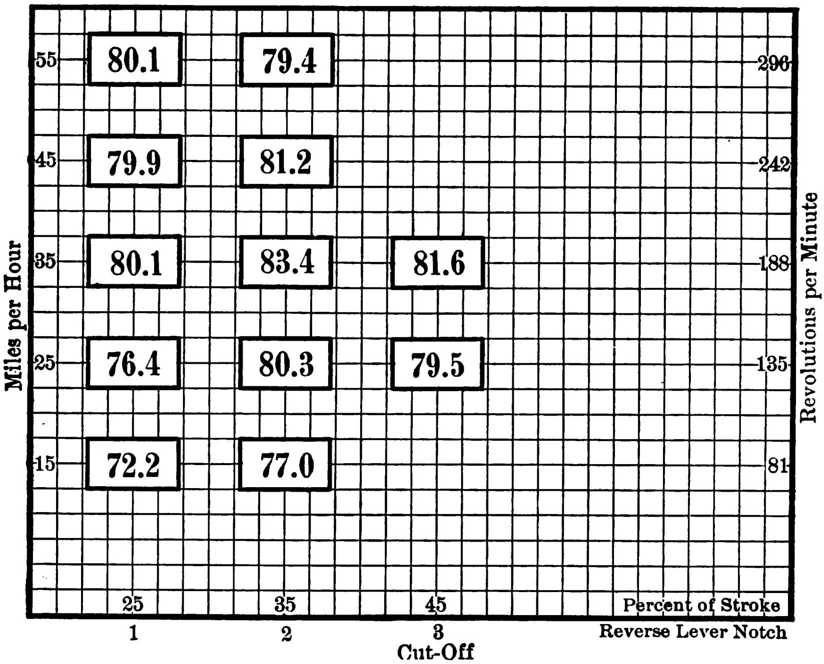


FIG. 68.—Percentage of Total Steam shown by Indicator at Cut-off.

were exhausted, leaving 0.30 of a pound during compression. When the speed is increased to 55 miles per hour, of the 0.88 of a pound in the cylinder, but 0.49, or a trifle more than half, is exhausted. It is, however, worthy of note that while the relative amount of mixture retained during compression is much greater at the higher speed, the actual increase is small since the weight of mixture during compression at the higher speed is but 0.39.

A more detailed exhibit of data derived from these tests, relating to the interchange of heat, will be found in Chapter IV., Series A.

TABLE XXIII.

THERMAL ACTION WITHIN THE CYLINDERS.

Designation of Test.	Weight of Mixture in Cylinder per Revolution. Lbs.	Weight of Mixture Exhausted per Revolution. Lbs.	Reëvaporation.		Condensation.	
			Per Revolution. Lbs.	Per I.H.P. per Hour. Lbs.	Per Revolution. Lbs.	Per I.H.P. per Hour. Lbs.
15-1-A	1.4166	1.1369	0.0106	0.3994		
25-1-A	1.0834	0.7816	0.0148	0.5307		
35-1-A	1.0456	0.7071	0.0182	0.7071		
45-1-A	0.9465	0.5787	0.0118	0.5859		
55-1-A	0.8848	0.4862	0.0237	1.4929		
15-2-A	1.7828	1.5692	......	......	0.0108	0.1903
25-2-A	1.5006	1.2555	0.0045	0.0961		
35-2-A	1.3026	1.0016	0.0155	0.4063		
45-2-A	1.1818	0.8432	0.0146	0.4921		
55-2-A	1.1406	0.7636	0.0220	0.9233		
25-3-A	1.8863	1.6663	......	......	0.0167	0.2646
35-3-A	1.6626	1.3528	0.0081	0.1802		

40. Boiler Performance.—Since the performance of the boiler is fully treated in another portion of this work, it is unnecessary to present an elaborate discussion in this connection. It will, however, serve in the further discussion of the performance of the locomotive, if some attention be given the limitation upon its capacity. This is well set forth in Fig. 69. The values of this figure show the pounds of water from and at 212 degrees, which were required to be evaporated each hour to sustain the operation of the locomotive under the conditions of speed and cut-off indicated. The values on shaded background represent an hourly evaporation of more than 12,500 pounds, and correspond to a rate of power which approaches closely to the maximum of the boiler when fired with Indiana block coal. With a better fuel it is probable that a total of 18,000 can be evaporated, an amount which is equal to 15 pounds of water per foot of heating surface per hour. Accepting this latter value as representing the maximum capacity of the boiler, then the points of cut-off and speed, for which the boiler will serve to meet the demands of the cylinders under a fully open throttle, will be represented by the curve in Fig. 69. Thus, at 25 miles steam can be supplied for a cut-off of over 55 per cent; while at a speed of 50 miles the cut-off cannot be much greater than 45 per cent. An attempt to employ longer cut-offs will result in failure through lack of steam. This curve

represents maximum performance, under a wide-open throttle, for the engine in question. It shows, also, the effect of changes in speed upon the total consumption. Thus, comparing tests for which the cut-off is the same, it will be seen that the demand upon the boiler is not greatly increased by increasing the speed. The reason for this has already been explained, but it will be of interest to note that the upper portion of the curve of maximum performance is so nearly vertical that, after the speed has reached 25 miles, but slight reduc-

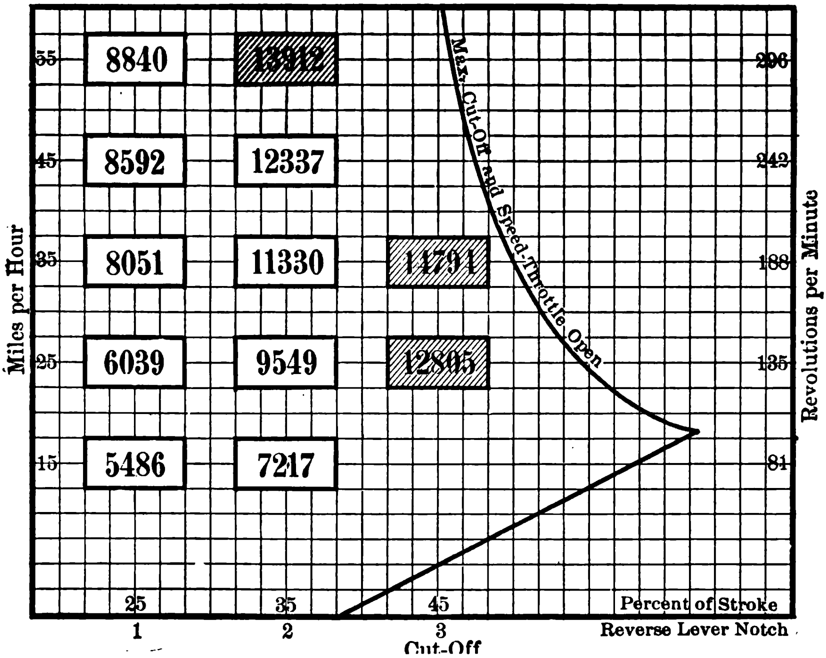


FIG. 69.—Equivalent Evaporation per Hour.

tions in the cut-off are required to enable the boiler to meet the demands of the cylinders under further increments of speed. Doubling the speed from 25 miles to 50 miles only requires the cut-off to be reduced from 55 per cent to 45 per cent; whereas, if it were not for the wire-drawing action, it might be expected that doubling the speed would necessitate the use of half the cut-off.

The pounds of water evaporated per pound of coal, as actually obtained for the tests, are given by the points in Fig. 70, and, as rectified by the use of the equation

$$E = 10.08 - .296H.$$

by the line in the same figure. A full statement of observed and calculated data, derived from the boiler as a result of the twelve tests under consideration, will be found in the recorded data, Chapter IV.

41. **Performance of the Locomotive as a Whole.** — Dealing first with the locomotive as a power-plant, consisting of boiler and engines, its performance is briefly set forth by Table XXIV. and Fig. 71. From these exhibits it appears that an indicated horse-power is under most conditions of operation obtained on a consump-

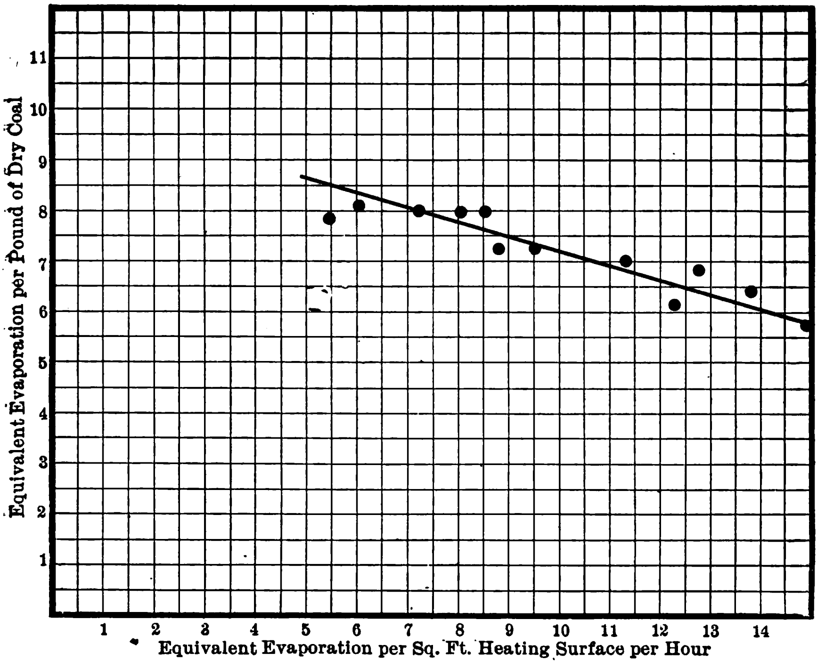


FIG. 70.—Evaporative Efficiency.

tion of from 4 to 6 pounds of Brazil block coal. With higher grades of coal it should be possible to lower the record by about one-fifth, making the range 3.2 to 4.8 pounds. These latter values may fairly be used in comparing the performance of a simple locomotive operating under ordinary conditions with steam-plants of other types. The rate at which power is developed is well shown by the table Each foot of heating surface in the boiler yields from .15 to .47 of a horse-power, and each foot of grate surface from 11 to 28 horse-power. Results which will approach these in value are not to be found in any other class of service.

TABLE XXIV.

BOILER AND ENGINE PERFORMANCE.

Designation of Test.	Coal per I.H.P. per Hour. Lbs.	Indicated Horse-power	
		Per Square Foot of Heating Surface.	Per Square Foot of Grate Surface.
15-1-A	4.16	0.157	10.90
25-1-A	4.05	0.178	12.40
35-1-A	4.19	0.248	17.23
45-1-A	4.52	0.249	17.27
55-1-A	4.87	0.239	16.63
15-2-A	4.16	0.226	14.99
25-2-A	4.40	0.297	20.65
35-2-A	4.67	0.357	24.87
45-2-A	5.26	0.359	25.01
55-2-A	6.40	0.361	25.03
25-3-A	5.41	0.370	25.80
35-3-A	6.25	0.407	28.31

TABLE XXV.

PERFORMANCE AT THE DRAW-BAR.

Designation of Test.	Steam Used per D.H.P. per Hour. Lbs.	Dry Coal Used per D.H.P. per Hour. Lbs.	Dynamometer Horse-power	
			Per Square Foot of Heating Surface.	Per Square Foot of Grate Surface.
15-1-A	32.92	4.69	0.138	9.7
25-1-A	33.87	4.90	0.148	10.4
35-1-A	32.93	5.14	0.203	14.3
45-1-A	37.45	5.95	0.190	13.4
55-1-A	43.86	7.01	0.167	11.8
15-2-A	29.64	4.48	0.202	14.2
25-2-A	28.91	4.77	0.274	19.3
35-2-A	29.18	5.19	0.322	22.7
45-2-A	32.59	6.06	0.315	22.1
55-2-A	38.03	7.59	0.304	21.3
25-3-A	30.27	5.77	0.351	24.7
35-3-A	32.44	6.79	0.379	26.7

Between the cylinders and the draw-bar there is, of course, some loss of power (Chapter XIX.), which, in the case of a locomotive on a testing-plant, amounts only to the friction of the machine. Since, however, it is the whole purpose of a locomotive to exert force at

the draw-bar, it is of importance to possess some measure of its performance in terms of force exerted at the draw-bar. Table XXV. gives a record of the weight of steam and coal used per hour in developing a horse-power at the draw-bar. The coal used is also shown diagrammatically by Fig. 72. The table gives also the horse-power developed at the draw-bar (dynamometer horse-power) per foot of heating surface and per foot of grate surface respectively.

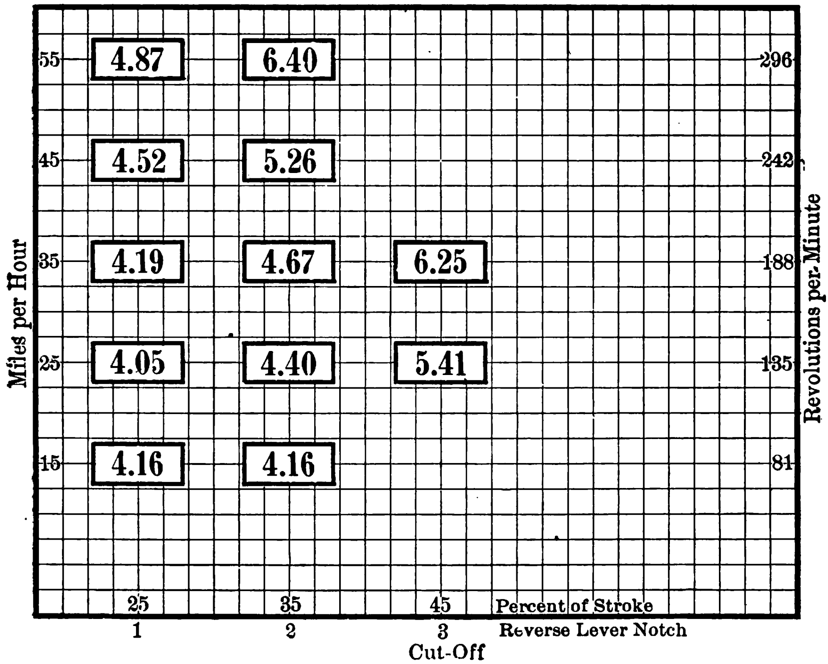


FIG. 71.—Coal per I.H.P. per Hour.

The effective pull exerted at the draw-bar is shown by Fig. 73 and the equivalent horse-power by Fig. 74. The horse-power equivalent to machine friction is shown by Fig. 75. A correct understanding of the general effect of increments of speed upon the forces exerted at the draw-bar is a matter of great importance to any one interested in locomotive performance. The effects are so well shown by the figures referred to as to make them well worthy of careful study. The coal used per mile run is shown by Fig. 76.

42. Maximum Power Dependent upon Efficiency.—It has already been stated that anything which operates to increase the

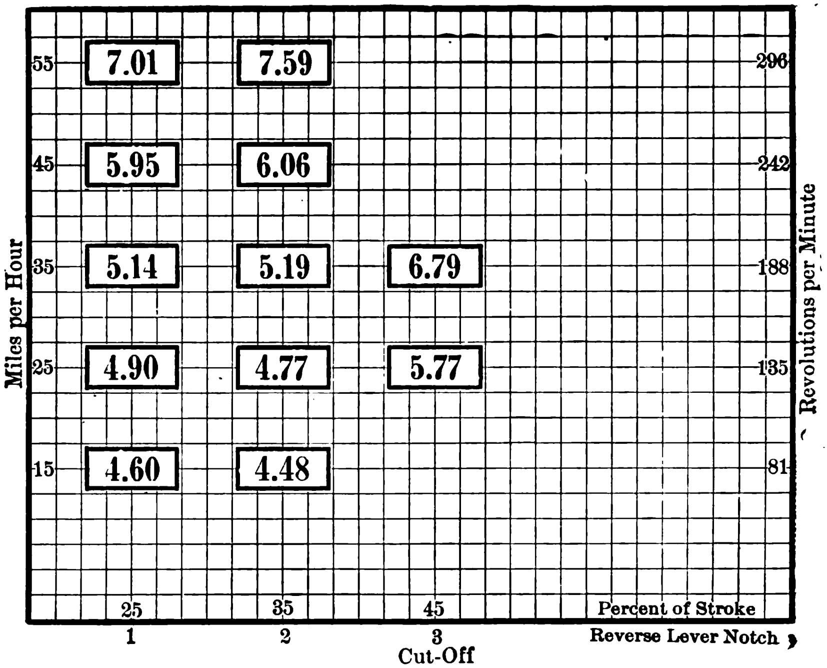


FIG. 72.—Coal per D.H.P. per Hour.

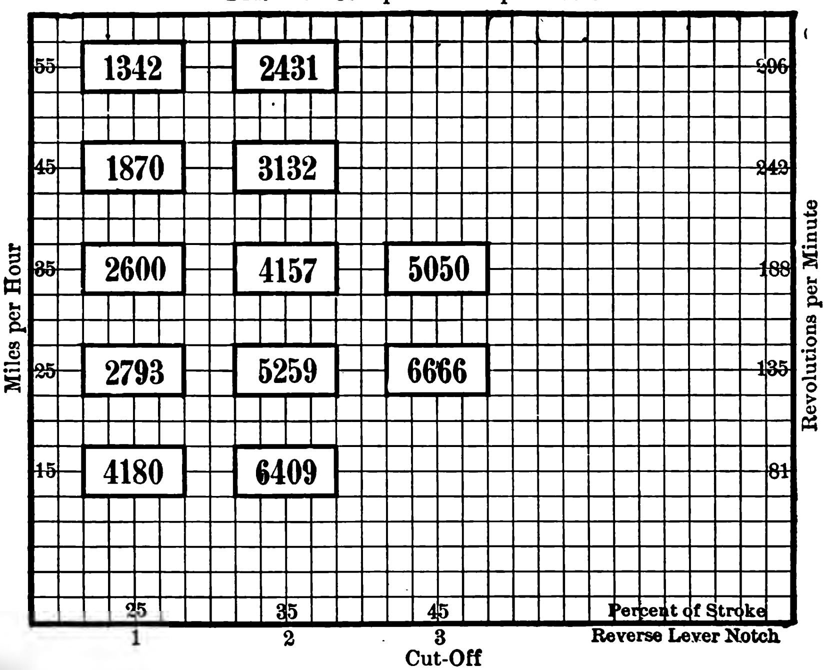


FIG. 73.—Draw-bar Pull.

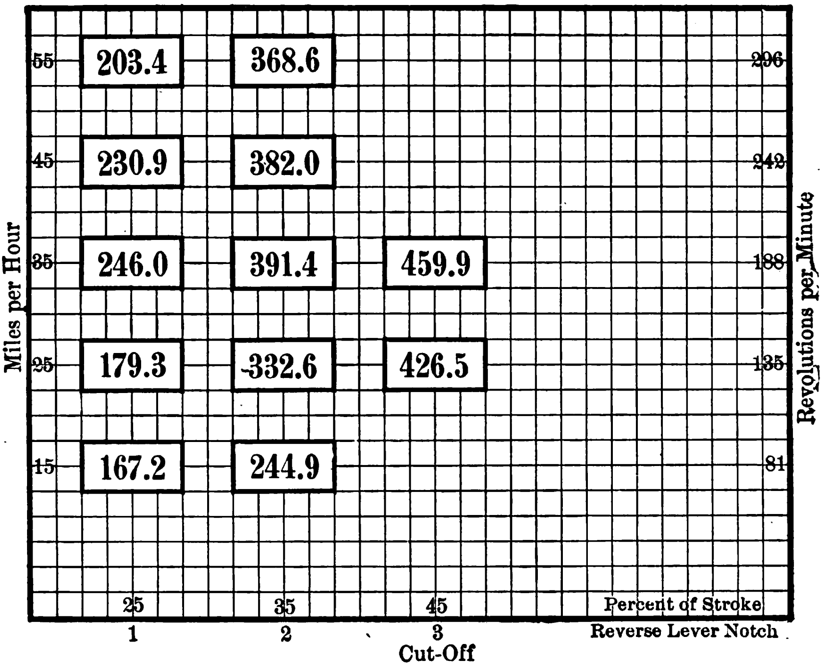


FIG. 74.—Dynamometer Horse-power.

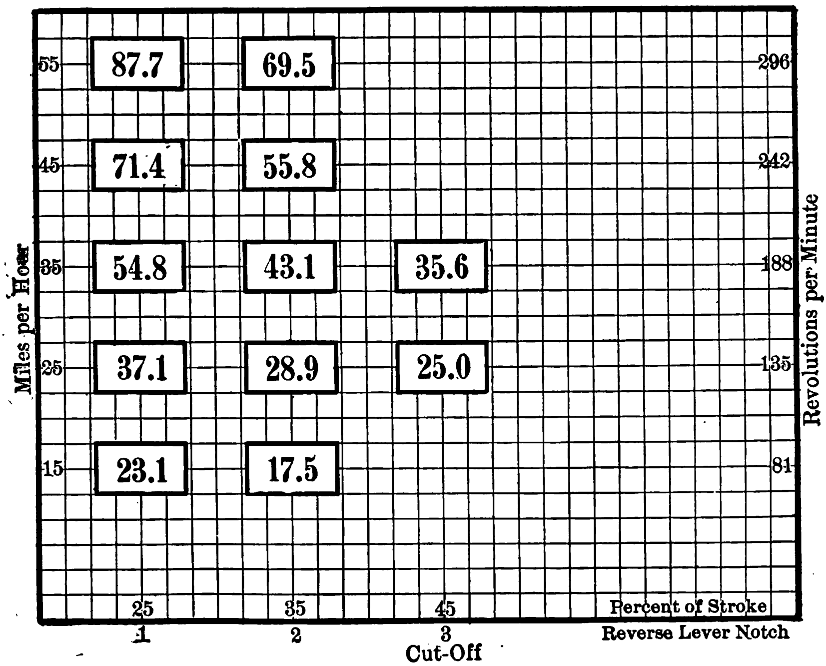


FIG. 75.—Friction Horse-power.

efficiency of a locomotive may, at the normal limit of power, be utilized in the development of more power. The importance of this statement entitles it to some further consideration.

It is evident that the maintenance of pressure in the cylinders demands steam from the boiler, and, hence, that the limit of cylinder work is reached when the boiler can no longer meet the demand which is made upon it. If now some improvement in fire-box or

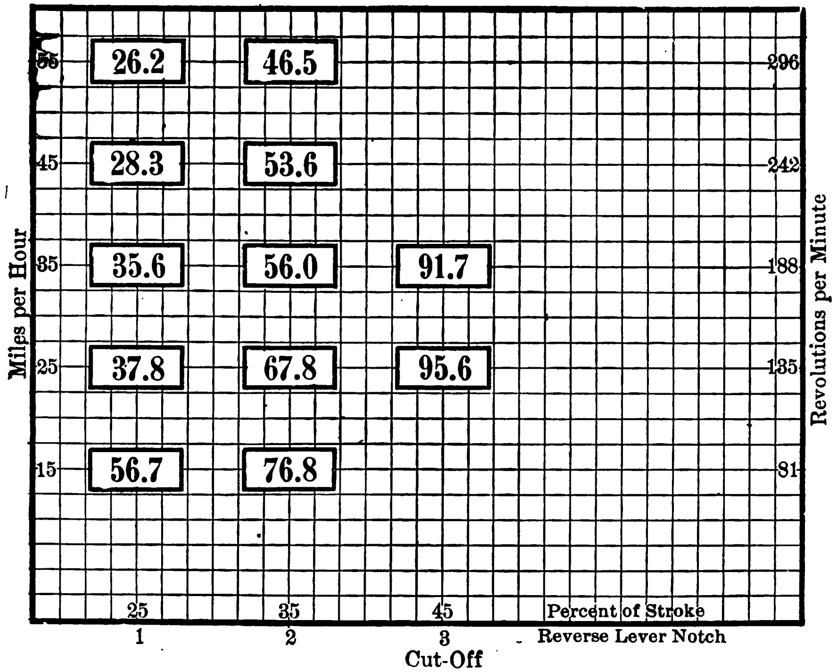


FIG. 76.—Coal per Mile Run.

boiler construction is adopted which will increase the evaporative efficiency of the boiler, making it possible to evaporate more water in return for each pound of fuel burned, then, at the sacrifice of the saving, it will become possible to establish a new and a higher limit in the capacity of the boiler. The more efficient boiler can be made to deliver more steam, which, in turn, may be utilized in the development of more power.

Again, assuming the boiler performance to remain unchanged, an improvement in the cylinder action will be found to give a similar result. For example, with a definite maximum quantity of

steam, the cylinders under normal conditions can deliver a certain maximum power. If now, by the adoption of compound cylinders, or of some other improvement, an arrangement can be had which delivers a horse-power upon the consumption of less steam than before, then the definite maximum quantity of steam which can be delivered by the boiler will not be entirely consumed in developing the normal cylinder power, and the surplus can be employed in developing more power.

In a similar manner it can be shown that any improvement which will increase the efficiency of any essential detail of the machine establishes a credit which, when the locomotive is pushed to its highest performance, may be invested in an increase of power.

There is, therefore, a twofold purpose to be served by any device or arrangement which adds to the efficiency of any important element of the locomotive: first, under normal conditions of operation, that of saving fuel, and, second, when the need is for the largest possible output of work, that of raising the maximum output of power.

III. THE BOILER.

CHAPTER VI.

BOILER PERFORMANCE.*

43. Selection of Data.—The tabulated data in Chapter IV. constitute the results obtained from forty-four efficiency tests of the experimental locomotive. It will be noted that the first thirty-five tests were all made within a period of two years between November 23, 1894, and February 11, 1897. The discussion in this chapter is based on these thirty-five tests. Other tests were made earlier and later than these, and some of the earlier results were published;† but inasmuch as the earlier results were for the most part obtained at a time when the mounting mechanism of the testing-plant was being perfected, and when the accessory apparatus for detecting errors was limited, they do not possess the same degree of reliability as those chosen for the present purpose. During the early existence of the plant the conditions affecting the operation of the boiler could not be maintained with the same degree of constancy as was afterward possible, and, to add to the irregularities, different men were employed in firing, which as a study of the results has shown, introduced a personal variable. The results of a considerable number of later tests were destroyed by fire.

The analysis which follows will show that the experimental results are in most respects consistent. Some of the facts which at first appear to be irregular will be accounted for, while others

* This discussion was presented as a paper before the American Society of Mechanical Engineers in December, 1900, forming part of Vol. XXII of the Transactions.

† Tests of the Locomotive at the Laboratory of Purdue University, Trans. M. E., Vol. XIV.

must be passed over as at present impossible of explanation. It is probable, however, that a full knowledge of all the conditions affecting the performance of the locomotive boiler may serve to make clear those phenomena which are at present inexplicable.

44. The Boiler.—A complete description of the boiler under consideration, together with a statement of all dimensions, will be found presented in Chapter III. The boiler, during its stay in the laboratory, suffered little or no change of condition. The water used carries some solid matter in solution, but it does not form hard scales, unless it is allowed to remain in the boiler for a considerable period. The boiler of Schenectady was never neglected, and at the close of its service in the laboratory it was clean and otherwise in as good condition as when work was commenced upon it.

45. General Conditions (Table XXVI).—To those accustomed to reviewing data derived from stationary boilers, the duration of the tests (Column 4) will appear insufficient. The arguments sustaining the practice involving short tests for locomotives on a testing-plant were presented to the committee having in charge the revision of the American Society of Mechanical Engineers' code relative to a standard method of conducting boiler-tests, and will be found in the published correspondence of that committee.* The tests herein recorded were run several years in advance of the presentation of the committee's final report, and for this reason it will be of interest to call attention to the fact that fifteen out of the thirty-five tests perfectly fulfill the requirements of the present code. In defense of those which are shorter than allowed by the code, it should be said that the writer's opinion on the subject was embodied in a recommendation to the committee to the effect that the limit applying in such cases be the burning of a total for the test of not less than 100 pounds of coal per foot of grate surface. The committee adopted the general form of the recommendation, but fixed the limit at 250 pounds. For two of the tests presented the total fuel per foot of grate is slightly below 150 pounds, and for four others it is below 200.

The boiler pressure (Column 5) was practically the same for all tests save two, and for these it was intentionally higher than the normal in the one case and lower in the other. Pressures were observed from an ordinary dial-gauge at five-minute intervals, and also recorded

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* Proceedings Am. Soc. Mech. Eng'rs, Vol. XXI, p. 112.

TABLE XXVI.

GENERAL CONDITIONS.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.			Duration of Test in Minutes.	Average Pressure, Pounds per Square Inch.			Average Temp., Deg. Fahr.	
Consecutive Number.	Laboratory Designation.	Date of Test.		Steam Pressure in Boiler by Gauge.	Atmospheric Pressure.	Absolute Steam Pressure.	Of Laboratory.	Of Feed-water.
1	2	3	4	5	6	7	8	9
1	15-1-A	Dec. 12, 1894	240	125.9	14.5	140.4	65	53.2
2	$15\text{-}1_b\text{-}V$	Nov. 23, 1894	190	127.3	14.5	141.8	74	53.8
3	15-1-H	Dec. 9, 1896	180	123.6	14.3	137.9	72	54.6
4	25-1-A	Dec. 14, 1894	255	120.1	14.5	134.6	67	53.8
5	15-1 G	Nov. 9, 1896	180	123.6	14.4	138.0	69	53.3
6	15-2-A	Nov. 13, 1895	180	129.5	14.4	143.9	79	55.2
7	25-1-V	Nov. 26, 1894	240	127.1	14.3	141.4	70	53.9
8	35-1-A	Dec. 17, 1894	180	129.7	14.6	144.3	75	53.2
9	35-2-F	Jan. 21, 1895	180	155.4	14.7	170.1	71	52.7
10	35-1-V	Dec. 7, 1894	140	128.9	14.3	143.2	69	51.9
11	35-2-E	Jan. 16, 1895	180	128.1	14.5	142.6	71	51.9
12	45-1-A	Nov. 20, 1895	150	128.8	14.3	143.1	76	56.4
13	35-2-B	Jan. 14, 1895	170	98,4	14.4	112.8	66	50.0
14	55-1-A	Nov. 25, 1895	120	124.9	14.2	139.1	76	56.0
15	35-1-H	Dec. 18, 1896	160	121.2	14.6	135.8	67	52.6
16	35-1-G	Nov. 20, 1896	170	125.0	14.6	139.6	74	55.0
17	$35\text{-}1_b\text{-}G$	Dec. 2, 1896	170	128.2	14.6	142.8	78	52.5
18	25-2-A	Oct. 25, 1895	180	129.3	14.4	143.7	85	56.0
19	55-1-V	Dec. 18, 1895	120	128.4	14.4	142.8	76	58.1
20	$35\text{-}2_b\text{-}H$	Feb. 10, 1897	160	122.6	14.5	137.1	74	50.6
21	55-1-H	Feb. 11, 1897	120	127.5	14.4	141.9	71	52.1
22	55-1-G	Nov. 23, 1896	120	126.9	14.5	141.4	79	54.2
23	15-9-A	Nov. 6, 1896	150	122.5	14.3	136.8	79	55.6
24	15-9-H	Dec. 11, 1896	180	122.7	14.4	137.1	81	53.2
25	35-2-A	Dec. 19, 1894	180	123.0	14.5	135.7	72	52.5
26	35-3-G	Dec. 4, 1896	140	125.9	14.3	140.2	70	52.6
27	15-9-G	Nov. 12, 1896	160	124.1	14.5	138.6	77	53.1
28	35-3-H	Dec. 14, 1896	120	116.5	14.3	130.8	73	53.0
29	35-2-G	Nov. 13, 1896	160	121.0	14.5	135.5	78	53.6
30	35-2-H	Dec. 16, 1896	120	112.0	14.6	126.6	72	53.0
31	45-2-A	Nov. 18, 1895	140	126.7	14.3	141.0	84	55.0
32	25-3-A	Nov. 1, 1895	122.5	127.2	14.5	141.7	76	53.3
33	35-2-C	Jan. 23, 1895	120	143.3	14.4	157.7	71	51.7
34	55-2-A	Nov. 22, 1895	68	124.0	14.4	138.4	76	58.4
35	35-3-A	Nov. 15, 1895	120	125.3	14.3	139.6	77	55.5

by a Bristol gauge. To better show the fluctuations in pressure, the chart of this and other recording-gauges used about the locomotive had a rapid motion, making a complete revolution in six hours. A sample chart is presented as Fig. 77.

46. Actual Evaporation (Table XXVII). — This table shows (Columns 10 and 11) the regularity with which water was delivered

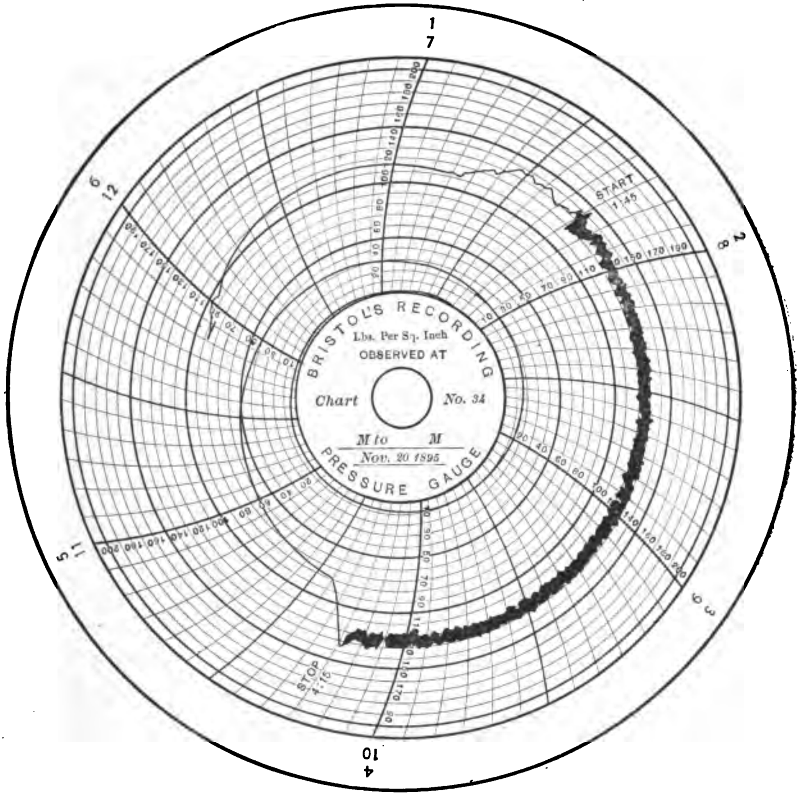

FIG. 77.—Chart from Recording-gauge showing Boiler Pressure.

to the boiler; the total pounds delivered to the injector during the test (Column 12); the pounds caught from injector overflow (Column 13); the pounds received by the boiler (Column 14); and the rate at which the evaporation proceeded (Column 15). In all tests an effort was made to keep the injector constantly in action, but for those of low power the rate of delivery could not be made sufficiently small to permit this being done. The data for a few of the tests show the injector to have been started two or more times, while its

Table XXVII.

ACTUAL EVAPORATION.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.			Water and Steam.					
Consecutive Number.	Laboratory Designation.	Duration of Test in Minutes.	Number of Times Injector was Started.	Minutes One or Both Injectors were in Action.	Total Pounds of Water Delivered to Injectors.	Pounds of Water Lost from Injector Overflow.	Pounds of Water Delivered to Boiler and Presumably Evaporated.	Pounds of Water Evaporated per Hour.
1	2	4	10	11	12	13	14	15
1	15-1-A	240	22	130	22,415	315	22,100	5,525
2	15-1b-V	190	14	129	17,925	135	17,790	5,618
3	15-1-H	180	7	126	18,349	126	18,223	6,075
4	25-1-A	255	10	236	25,995	100	25,895	6,093
5	15-1-G	180	12	127	18,768	72	18.696	6.232
6	15-2-A	180	9	149	21,885	44	21,841	7,280
7	25-1-V	240	7	213	29,508	32	29,476	7,369
8	35-1-A	180	2	180	24,468	100	24,368	8,123
9	35-2-F	180	9	149	24,793	460	24,333	8,111
10	35-1-V	140	1	136	19,899	0	19,899	8,528
11	35-2-E	180	3	175	25,742	45	25,697	8,566
12	45-1-A	150	2	146	21,688	15	21,673	8,669
13	35-2-B	170	8	158	24,728	173	24,555	8,666
14	55-1-A	120	1	120	17,970	87	17,883	8,941
15	35-1-H	160	1	160	24,793	70	24,723	9,271
16	35-1-G	170	1	170	26.598	35	26,563	9,375
17	35-1b-G	170	1	170	26,753	10	26,743	9.439
18	25-2-A	180	1	180	28,911	5	28,906	9,635
19	55-1-V	120	1	120	19,526	108	19,418	9.709
20	35-2b-H	160	1	160	25,834	5	25,829	9,686
21	55-1-H	120	1	120	20,514	0	20,514	10,257
22	55-1-G	120	1	120	20.791	30	20,761	10,381
23	15-9 A	150	1	150	28,582	60	28,522	11,409
24	15-9-H	180	1	180	34,403	30	34,373	11,458
25	35-2-A	180	1	180	34,354	29	34,325	11,442
26	35-3-G	140	1	140	27,060	0	27.060	11,597
27	15-9-G	160	1	154	30,877	5	30.872	11,577
28	35-3-H	120	1	120	23,446	60	23,386	11,693
29	35-2-G	160	1	160	32,886	40	32,846	12,317
30	35-2-H	120	3	115	24,793	60	24,733	12,367
31	45-2-A	140	1	140	29,313	213	29,100	12,472
32	25-3-A	122	1	122	26,415	14	26,401	12,931
33	35-2-C	120	3	120	26,034	25	26,009	13,004
34	55-2-A	68	1	68	16,188	277	15,911	14,039
35	35-3-A	120	1	120	29,984	109	29,875	14,937

period of action is recorded as coincident with the length of the test. This apparent inconsistency is explained by the fact that it was sometimes convenient to change from one injector to the other during the progress of the tests, and at other times both injectors were in action at the same time. Column 10 merely shows the number of starts made, and includes the record for both injectors. The last column of this table discloses something of the significance of the conditions attending the action of the locomotive boiler. For example, test 35 shows that nearly 15,000 pounds of water were delivered to the boiler and presumably evaporated each hour, or, approximately, 250 pounds per minute. This rate of very nearly a barrel a minute is sufficient to evaporate an amount of water equal to the full water capacity of the boiler in 34 minutes. At this rate, had the injectors ceased in their action, the water level would have fallen between the upper and the middle gauges at the rate of one inch each minute.

47. **Quality of Steam and Equivalent Evaporation (Table XXVIII).**—The quality of steam, assuming dry saturated steam to be unity, is shown by Column 16, and the percentage of moisture by Column 17. Results were obtained by the use of a throttling calorimeter of an approved form, taking steam from a perforated pipe extending horizontally into the dome of the boiler at a point *A*, Fig. 35 (Chapter III.), near the throttle-opening. Observations were made at five-minute intervals. An examination of the table will show that, in general, the amount of moisture in the steam increases as the rate of evaporation is increased, though variations in individual results are so great that they do not fall in any well-defined line. The reason for apparent inconsistencies is to be looked for either in the methods employed, or in actual variations in the performance of the boiler. The methods employed were the same for all tests, while the condition of the boiler was necessarily subject to change. The boiler had to be frequently washed. It is conceivable that dryer steam may be furnished by the boiler when newly washed than when in a condition requiring washing, though there is nothing in the data either to sustain or to discredit such a conclusion. It has, however, been shown by Professor Jacobus * that almost all moisture which may be intermixed with steam passing a horizontal pipe separates itself

* Tests to show the Distribution of Moisture in Steam when flowing in a Horizontal Pipe, Transactions of the American Society of Mechanical Engineers, Vol. XVI, p 1017.

TABLE XXVIII.

QUALITY OF STEAM AND EQUIVALENT EVAPORATION.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.		Duration of Test in Minutes.	Results of Calorimeter Tests.		Equivalent Evaporation from and at 212° Fahr. per Hour.	
Consecutive Number.	Laboratory Designation.		Quality of Steam in Dome of Boiler, Dry, Saturated Steam being Taken as Unity.	Percentage of Moisture in Steam.	Assuming Quality of Steam as Shown by Column 16.	Assuming all Water Delivered to Boiler to have been Completely Evaporated into Dry Steam.
1	2	4	16	17	18	19
1	15-1-A	240	.9951	0.49	6,659	6,683
2	15-1$_b$-V	190	.9937	0.63	6,763	6,794
3	15-1-H	180	.9838	1.62	7,249	7,337
4	25-1-A	255	.9922	0.78	7,318	7,360
5	15-1-G	180	.9894	1.06	7,476	7,535
6	15-2-A	180	.9924	0.76	8,745	8,796
7	25-1-V	240	.9932	0.68	8,865	8,910
8	35-1-A	180	.9917	0.83	9,771	9,832
9	35-2-F	180	.9900	1.00	9,785	9,855
10	35-1-V	140	.9937	0.63	10,284	10,332
11	35-2-E	180	.9932	0.68	10,324	10,377
12	45-1-A	150	.9913	0.87	10,395	10,463
13	35-2-B	170	.9931	0.69	10,413	10,467
14	55-1-A	120	.9878	1.22	10,691	10,788
15	35-1-H	160	.9851	1.49	11.090	11,214
16	35-1-G	170	.9886	1.14	11,226	11,322
17	35-1$_b$-G	170	.9860	1.40	11,312	11,430
18	25-2-A	180	.9910	0.90	11,554	11,633
19	55-1-V	120	.9912	0.88	11,624	11,700
20	35-2$_b$-H	160	.9880	1.20	11,634	11,737
21	55-1-H	120	.9871	1.29	12,303	12,422
22	55-1-G	120	.9869	1.31	12,426	12,548
23	15-9-A	150	.9906	0.94	13,670	13,766
24	15-9-H	180	.9871	1.29	13,722	13,853
25	35-2-A	180	.9894	1.06	13,735	13,843
26	35-3-G	140	.9838	1.62	13,867	14,035
27	15-9-G	160	.9871	1.29	13,869	14,002
28	35-3-H	120	.9873	1.27	13,995	14,128
29	35-2-G	160	.9856	1.44	14,725	14,884
30	35-2-H	120	.9866	1.34	14,771	14,932
31	45-2-A	140	.9876	1.24	14,926	15,065
32	25-3-A	122.5	.9889	1.11	15,515	15,644
33	35-2-C	120	.9930	0.70	15,709	15,788
34	55-2-A	68	.9887	1.13	16,773	16,903
35	35-3-A	120	.9889	1.11	17,883	18,032

from the steam by gravitation, and forms a rill in the bottom of the pipe, the steam above being approximately dry. Experiments by the writer, involving a visual examination of the steam-space of a boiler while in action, revealed no haze or mist above the surface of the water, thus sustaining the conclusion that the steam within the steam-space of the boiler is ordinarily dry and saturated; the water is present as water and the steam as steam. They are not intermixed. If this is true, the steam which passes the throttle of a locomotive should be expected to be dry, and it would be entirely so if it were not that the violence of the circulation projects small beads of water upward, far beyond the general surface. Some of these enter the throttle with the steam. This action explains why the moisture increases with the power of the boiler, and makes it not unreasonable to assume that the purity of the water in the boiler may actually affect the quality of the steam.

The comparative dryness of the steam under all conditions is a fact worthy of emphasis, for the locomotive is often credited with carrying over a great deal of water to the cylinders. The tests show that this does not happen under constant conditions of running. When it occurs it is probably the result of too high a water level or of a sudden demand upon the boiler. For example, if the throttle of a locomotive, which has been for some time inactive, is quickly opened, large volumes of steam-bubbles leave the heating surface and crowd to the upper part of the boiler, making spray in the dome, a portion of which may pass out with the steam. A similar action occurs when an engine, which has been working at a light load, is suddenly required to increase its power. But these are exceptional conditions. Under uniform conditions of running, such as prevailed throughout the tests herein presented, the moisture passing the throttle is never great.

If, as the writer believes, it is fair to conclude that variations in moisture are largely due to incidental conditions, no serious mistake would be made if the indications of the calorimeter were entirely disregarded, and all calculations based on the assumption that the steam generated is dry and saturated. Results thus obtained should be somewhat more consistent, one with another, than those corrected for moisture, and hence for general purposes more satisfactory. In accord with this view, while none of the calorimeter work has been ignored in calculating results, many of the derived results have been carried out in duplicate. Thus, the equivalent evaporation is first

determined on the assumption that the steam has the quality shown by the calorimeter (Column 18), and is also determined on the assumption that all water delivered to the boiler is evaporated into dry and saturated steam (Column 19). In accord with the usual practice, however, comparisons which follow are based on the corrected results (Column 18).

48. Power of Boiler (Table XXIX).—The power developed by the boiler is proportional to the rate of evaporation (Column 18). A few comparisons will serve to show something of the peculiar conditions under which the boiler of a locomotive performs its service. Thus, the water evaporated per foot of grate surface (Column 20) varies from less than 400 to more than 1,000 pounds per hour. These figures reflect well the intensity of the furnace action, which must provide for the combustion of sufficient fuel to produce such a result. Similarly, the weight of water evaporated per square foot of heating surface (Column 21) varies from 5½ to almost 15 pounds per hour, the maximum rate being nearly the equivalent of a boiler horse-power for every 2 square feet of heating surface in the boiler. The total boiler horse-power (Column 22), the horse-power per square foot of heating surface (Column 23), and the horse-power pĕr square foot of grate surface (Column 24) are also of interest, especially for the higher power tests.

49. Coal and Combustible (Table XXX).—Attention has been called to the very large amount of water evaporated by the boiler tested. It follows that a correspondingly large amount of fuel is needed to bring about this evaporation. The record appears in Table XXX.

The coal used for all tests was Indiana block, mined in the neighborhood of Brazil. It burns to a white ash, without clinkers, and is light and friable, which qualities prevent its giving maximum results in locomotive service. The composition of several representative samples proved to be as follows:

	1	2	3	4
Per cent fixed carbon	49.65	51.84	51.09	51.59
Per cent volatile matter	40.29	39.00	38.93	38.87
Per cent combined moisture	3.15	3.62	2.35	3.44
Per cent ash	6.91	5.54	7.63	6.10
	100.00	100.00	100.00	100.00

In each test the coal was weighed—a barrowful at a time—as it was dumped at the feet of the fireman. A sample of from 50 to 100

TABLE XXIX.

POWER.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.		Duration of Test in Minutes.	Equivalent Evaporation from and at 212° Fahr. Pounds, assuming Quality of Steam as shown by Column 16.			Rated Horse-power 34.5 Evaporative Units, assuming Quality of Steam as shown by Column 16.		
Consecutive Number.	Laboratory Designation.		Water per Hour.	Water Evaporated per Square Foot of Grate Surface per Hour.	Water Evaporated per Square Foot of Heating Surface per Hour.	Total.	Per Square Foot of Heating Surface.	Per Square Foot of Grate Surface.
1	2	4	18	20	21	22	23	24
1	15-1-A	240	6,659	386	5.48	193	.159	11.2
2	15-1b-V	190	6,763	392	5.57	196	.161	11.4
3	15-1-H	180	7,249	420	5.97	210	.173	12.2
4	25-1-A	255	7,318	424	6.03	212	.175	12.3
5	15-1-G	180	7,476	433	6.16	217	.179	12.6
6	15-2-A	180	8,745	507	7.20	253	.208	14.7
7	25-1-V	240	8,865	514	7.30	257	.212	14.8
8	35-1-A	180	9,771	566	8.05	283	.233	16.4
9	35-2-F	180	9,785	567	8.06	284	.234	16.4
10	35-1-V	140	10,284	596	8.47	298	.245	17.3
11	35-2-E	180	10,324	598	8.51	299	.246	17.3
12	45-1-A	150	10,395	603	8.56	301	.248	17.4
13	35-2-B	170	10,413	604	8.57	302	.249	17.5
14	55-1-A	120	10,691	619	8.80	310	.255	17.9
15	35-1-H	160	11,090	643	9.13	321	.264	18.6
16	35-1-G	170	11,226	650	9.24	325	.268	18.8
17	35-1b-G	170	11,312	655	9.31	328	.270	19.0
18	25-2-A	180	11,554	669	9.51	335	.276	19.4
19	55-1-V	120	11,624	673	9.57	337	.278	19.5
20	35-2b-H	160	11,634	674	9.58	337	.278	19.5
21	55-1-H	120	12,303	713	10.13	357	.294	20.7
22	55-1-G	120	12,426	720	10.23	360	.296	20.9
23	15-9-A	150	13,670	792	11.25	396	.326	22.9
24	15-9-H	180	13,722	796	11.29	398	.328	23.1
25	35-2-A	180	13,735	796	11.31	398	.328	23.1
26	35-3-G	140	13,867	804	11.42	402	.331	23.3
27	15-9-G	160	13,869	804	11.42	402	.331	23.3
28	35-3-H	120	13,995	811	11.52	406	.334	23.5
29	35-2-G	160	14,725	853	12.12	427	.352	24.8
30	35-2-H	120	14,771	856	12.16	428	.353	24.8
31	45-2-A	140	14,926	865	12.29	433	.357	25.1
32	25-3-A	122.5	15,515	899	12.77	450	.370	26.1
33	35-2-C	120	15,709	911	12.93	455	.375	26.4
34	55-2-A	68	16,773	973	13.81	486	.400	28.2
35	35-3-A	120	17,883	1,037	14.93	518	.427	30.0

TABLE XXX.

COAL AND COMBUSTION.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.			Fuel: Indiana Brazil Block Coal, Pounds.					Rate of Combustion.	
Consecutive Number.	Laboratory Designation.	Duration of Test in Minutes.	Total Dry Coal Fired.	Total Refuse Caught in Ash-pan.	Total Combustible by Analysis.	Dry Coal Fired per Hour.	Combustible Fired per Hour.	Dry Coal Fired per Sq. Ft. of Grate Surface per Hour	Dry Coal Fired per Sq. Ft. of Heating Surface per Hour
1	2	4	27	28	29	30	31	32	33
1	15–1–A	240	3,399	292	3,059	850	765	49.3	.700
2	15–1$_b$–V	190	2,577	261	2,319	814	732	47.2	.670
3	15–1–H	180	2,375	240	2,138	792	713	45.9	.652
4	25–1–A	255	3,864	93	3,478	909	818	52.8	.748
5	15–1–G	180	2,678	262	2,410	893	803	51.8	.735
6	15–2–A	180	3,297	258	2,967	1,099	989	63.7	.905
7	25–1–V	240	4,460	320	4,014	1,115	1,003	64.6	.918
8	35–1–A	180	3,785	299	3,406	1,262	1,135	73.1	1.038
9	35–2–F	180	3,859	295	3,473	1,286	1,158	74.6	1.059
10	35–1–V	140	3,180	272	2,862	1,363	1,227	78.9	1.122
11	35–2–E	180	4,107	353	3,696	1,369	1,232	79.3	1.127
12	45–1–A	150	3,272	231	2,945	1,309	1,178	75.9	1.078
13	35–2–B	170	4,370	461	3,933	1,542	1,388	89.4	1.270
14	55–1–A	120	2,980	220	2,682	1,490	1,341	86.4	1.227
15	35–1–H	160	4,152	320	3,737	1,557	1,401	90.3	1.283
16	35–1–G	170	4,288	342	3,859	1,513	1,362	87.7	1.246
17	35–1$_b$–G	170	4,150	380	3,735	1,465	1,318	84.9	1.206
18	25–2–A	180	4,826	25	4,343	1,609	1,448	93.3	1.324
19	55–1–V	120	3,330	239	2,997	1,665	1,498	96.5	1.371
20	35–2$_b$–H	160	4,321	332	3,889	1,620	1,458	93.9	1.334
21	55–1–H	120	3,203	214	2,883	1,602	1,441	92.8	1.319
22	55–1–G	120	3,769	241	3,392	1,885	1,696	109.2	1.552
23	15–9–A	150	5,014	265	4,513	2,006	1,805	116.3	1.651
24	15–9–H	180	6,363	344	5,727	2,121	1,909	123.0	1.746
25	35–2–A	180	5,933	258	5,339	1,978	1,780	114.6	1.628
26	35–3–G	140	4,708	365	4,237	2,018	1,816	117.0	1.662
27	15–9–G	160	5,487	354	4,938	2,058	1,852	119.3	1.694
28	35–3–H	120	4,248	346	3,823	2,124	1,912	123.2	1.749
29	35–2–G	160	6,183	363	5,565	2,319	2,087	134.4	1.909
30	35–2–H	120	4,250	297	3,823	2,125	1,912	123.2	1.750
31	45–2–A	140	5,720	243	5,148	2,452	2,206	142.1	2.019
32	25–3–A	122.5	4,684	91	4,216	2,294	2,065	133.0	1.889
33	35–2–C	120	5,356	384	4,820	2,678	2,410	155.3	2.205
34	55–2–A	68	2,995	205	2,643	2,695	2,378	153.2	2.176
35	35–3–A	120	6,266	298	5,639	3,133	2,819	181.6	2.581

pounds was put into a large galvanized iron pan, and air-dried. From results thus obtained, the total weight of coal fired was corrected for accidental moisture, giving results which appear in Column 27. As the coal was stored under roof, the correction was always small.

The total weight of dry coal fired, as given for the several tests (Column 27), contains two variables: the length of the test and the rate of combustion. The values given as dry coal fired per hour (Column 30) eliminate the first variable, and supply a true basis from which to compare the rates of combustion incident to the several tests. It will be seen that the amount of coal fired per hour is between the limits of 792 pounds and 3,133 pounds. Five tests have a rate of less than ½ a ton per hour, and twelve have a greater rate than 1 ton per hour. The rate per square foot of grate per hour (Column 32) ranges from 46 to 182, values the significance of which appears when it is considered that even in naval service, under forced draft, the rate seldom exceeds 60 pounds per hour. The coal burned per foot of heating surface per hour (Column 33) varies from .7 to 2.6 pounds.

The amount of refuse caught in the ash-pan (Column 28) is an item of no great importance in the case of locomotive boilers, since a large amount of non-combustible material which would, under the conditions of stationary practice, lodge in the ash-pan is, in locomotive service, thrown out at the top of the stack. The proportion of the whole amount of ash contained by the fuel which appears in the ash-pan depends upon the force of the draft, or, in other words, upon the rate of power at which the boiler is worked. It is greatest when the rate of combustion is least. When the engine is worked at very high power, the amount of refuse in the ash-pan, with the light fuel employed in the tests under consideration, is almost negligible.

In stationary practice the total combustible (Column 29) is obtained by subtracting the weight of refuse from the weight of coal. For reasons already explained, such a process gives no useful result when applied to the tests of locomotive practice. For the present purpose, therefore, resort has been had to the chemical analysis of the fuel, which shows about one-tenth the weight of the dry coal to be non-combustible. The total combustible is therefore assumed to equal nine-tenths of the weight of dry coal fired. The combustible per hour on this basis is shown as Column 31.

50. Thermal Units (Table XXXI).—The thermal units imparted to each pound of water passing the boiler, or

$$Q = xr + q - q_o,$$

Table XXXI.

THERMAL UNITS.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.		Duration of Test in Minutes.	British Thermal Units. Assuming Steam to have the Quality as shown by Column 16.			
Consecutive Number.	Laboratory Designation.		Per Pound of Steam Generated.	Total per Minute.	Per Pound of Dry Coal.	Per Pound of Combustible.
1	2	4	34	35	36	37
1	15–1–A	240	1,164.1	107,194	7,569	8,410
2	15–1b–V	190	1,162.6	108,858	8,026	8,918
3	15–1–H	180	1,152.4	116,668	8,842	9,823
4	25–1–A	255	1,160.0	117,798	7,774	8,638
5	15–1–G	180	1,158.6	120,340	8,088	8,988
6	15–2–A	180	1,160.2	140,778	7,686	8,541
7	25–1–V	240	1,161.8	142,688	7,678	8,531
8	35–1–A	180	1,161.7	157,268	7,479	8,310
9	35–2–F	180	1,165.1	157,502	7,348	8,163
10	35–1–V	140	1,164.7	165,546	7,288	8,098
11	35–2–E	180	1,164.1	166,188	7,283	8,093
12	45–1–A	150	1,158.1	167,336	7,671	8,523
13	35–2–B	170	1,160.5	167,624	6,521	7,245
14	55–1–A	120	1,154.7	172,079	6,729	7,699
15	35–1–H	160	1,155.3	178,516	6,879	7,643
16	35–1–G	170	1,156.5	180,706	7,164	7,961
17	35–1b–G	170	1,157.4	182,072	7,458	8,287
18	25–2–A	180	1,158.2	185,994	6,937	7,709
19	55–1–V	120	1,156.3	187,109	6,743	7,492
20	35–2b–H	160	1,160.0	187,260	6,934	7,704
21	55–1–H	120	1,158.5	198,046	7,419	8,243
22	55–1–G	120	1,156.1	200,015	6,369	7,076
23	15–9–A	150	1,157.2	220,038	6,583	7,314
24	15–9–H	180	1,156.6	220,866	6,248	6,942
25	35–2–A	180	1,159.3	221,092	6,707	7,452
26	35–3–G	140	1,154.8	223,206	6,637	7,375
27	15–9–G	160	1,157.0	223,243	6,510	7,234
28	35–3–H	120	1,155.9	225,266	6,363	7,071
29	35–2–G	160	1,154.6	237,025	6,134	6,815
30	35–2–H	120	1,153.5	237,746	6,713	7,459
31	45–2–A	140	1,155.9	240,262	5,881	6,534
32	25–3–A	122	1,158.8	249,743	6,531	7,256
33	35–2–C	120	1,166.7	252,873	5,666	6,296
34	55–2–A	68	1,153.9	269,996	6,130	6,812
35	35–3–A	120	1,156.3	287,871	5,513	6,126

are given in Column 34. The rate at which heat is transferred, as indicated by the thermal units absorbed by the water of the boiler each minute, is given in Column 35, while the thermal units absorbed

per pound of dry coal burned are given in Column 36, and per pound of combustible in Column 37.

No attempt has been made to express in numerical terms the thermal efficiency of the boiler. The determination of such a value depends upon the heating value of the fuel, which is not known in precise terms. It is probably not far from 13,000 thermal units per pound of dry coal. Comparing this value with the number of thermal units taken up by the water of the boiler for each pound of coal burned (Column 36), an approximate estimate of the thermal efficiency of the boiler may be had.

While the facts presented by this table are especially for the convenience of those who may desire to compare the performance of the boiler tested with data from other boilers, they are not without interest in themselves. For example, it is of interest to see that in test No. 35 the boiler transmitted approximately 288,000 thermal units per minute. That is, it delivered heat sufficient to raise the temperature of 144 tons of water one degree every minute. As many locomotives are now in service having more than double the power of the one tested, it may be said that the modern locomotive is capable of delivering sufficient heat to raise 300 tons of water one degree in temperature each minute.

51. Draft, Rate of Combustion, and Smoke-box Temperature (Table XXXII).—For the present purpose draft is defined as the difference between the pressure of the atmosphere and that of the smoke-box. The draft-gauge consists of a U tube partially filled with water, and securely attached to a pillar of the laboratory. One leg of the tube is in pipe connection with the interior of the smoke-box, the opening being at the point *C*, Fig. 35, and on the axis of the boiler. Observations were made at five-minute intervals. For the tests reported, the average draft (Column 38) varies from 1.7 inches to 7.5 inches.

In any boiler the condition of draft determines the rate of combustion, and consequently, under ideal conditions, the draft will be a function of the rate of combustion. But under conditions actually affecting the action of the boiler of a locomotive there are variations in this relationship. The precise action of the steam-jet in producing a draft action has been discussed in another place.* It is shown elsewhere in this discussion † that, other things

* Report of Committee on "Exhaust-pipes and Steam-passages." Proceedings of the American Railway Master Mechanics' Association, 1896.

† Chapter XI.

TABLE XXXII.

DRAFT, RATE OF COMBUSTION, AND SMOKE-BOX TEMPERATURE.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.		Duration of Test in Minutes.	Average Pressure. Draft, or Vacuum, in Smoke-box, Inches of Water.	Average Temperature, Deg. Fahr., of Smoke-box.	Fuel: Indiana Brazil Block Coal, Pounds. Dry Coal Fired per Hour.
Consecutive Number.	Laboratory Designation.				
1	2	4	38	39	40
1	15-1-A	240	1.72	553	850
2	15-1b-V	190	2.04	550	814
3	15-1-H	180	1.93	570	792
4	25-1-A	255	1.93	567	909
5	15-1-G	180	1.87	583	893
6	15-2-A	180	2.42	621	1,099
7	25-1-V	240	2.60	606	1,115
8	35-1-A	180	3.00	628	1,262
9	35-2-F	180	2.57	653	1,286
10	35-1-V	140	3.43	633	1,363
11	35-2-E	180	2.89	652	1,369
12	45-1-A	150	2.68	644	1,309
13	35-2-B	170	3.28	664	1,542
14	55-1-A	120	2.58	675	1,490
15	35-1-H	160	3.00	655	1,557
16	35-1-G	170	3.02	685	1,513
17	35-1b-G	170	2.98	618	1,465
18	25-2-A	180	3.37	696	1,609
19	55-1-V	120	3.20	667	1,665
20	35-2b-H	160	3.18	689	1,620
21	55-1-H	120	3.44	695	1,602
22	55-1-G	120	3.57	...	1,885
23	15-9-A	150	4.56	724	2,006
24	15-9-H	180	4.99	696	2,121
25	35-2-A	180	4.42	720	1,978
26	35-3-G	140	4.88	719	2,018
27	15-9-G	160	4.76	724	2,058
28	35-3-H	120	4.52	...	2,124
29	35-2-G	160	4.65	655	2,319
30	35-2-H	120	4.33	...	2,125
31	45-2-A	140	4.93	741	2,452
32	25-3-A	122.5	5.45	762	2,294
33	35-2-C	120	5.13	738	2,678
34	55-2-A	68	4.58	755	2,695
35	35-3-A	120	7.48	798	3,133

being equal, the capacity of the jet as a means for producing draft is nearly proportional to the weight of steam discharged per unit of time; and that, whether the discharge is in the slow, heavy puffs inci-

dent to slow speed, or in lighter but more rapid impulses, is not material. If the weight of steam discharged is the same, the draft action is approximately constant. The reduction of pressure in the smoke-box, however, which may result from the action of the

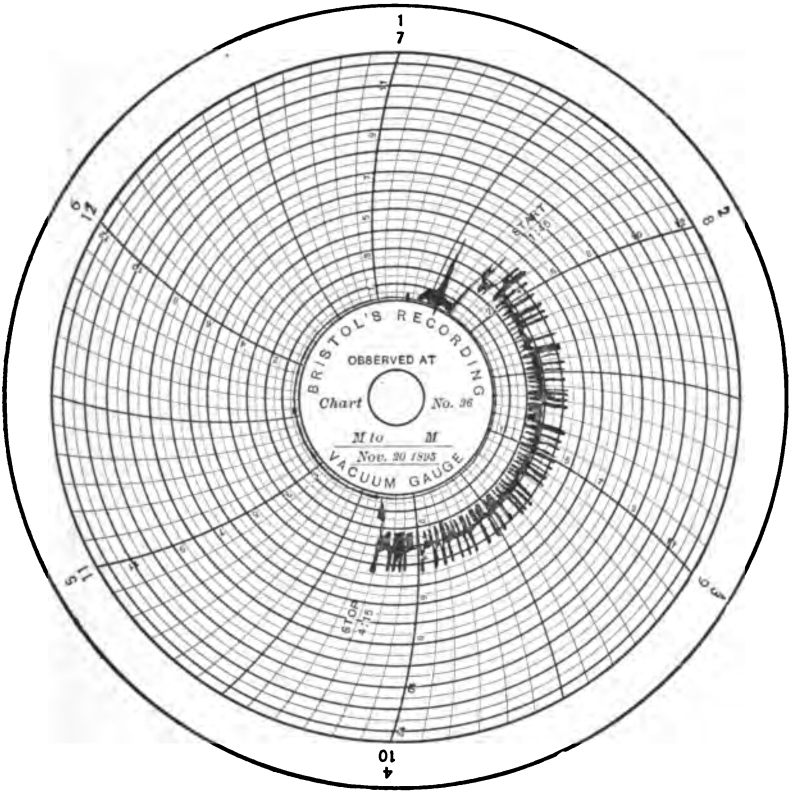

Fig. 78.—Chart from Recording-gauge showing Draft.

This and other charts from recording-gauges are from test No. 12. The test began with a flying start at 1.45. The normal draft from the test was a little less than 3 inches (exactly 2.68 inches). Where the line rises above this value it indicates that the dampers were closed, and where it falls below, that the fire-door was open. So far as the record shows the frequency with which the dampers were changed, it is an unusual one. Referring to the diagrams of other gauges (Figs. 77 and 79), the constancy of steam pressure and of draft may be judged.

exhaust under given conditions, depends upon the freedom with which air is permitted to pass into the fire-box. If the fire is thick and solid, the draft, as determined by the reduction of pressure in the smoke-box, will be high; if the fire is light, the draft will

be low. This is well shown by Fig. 78, which is a representative diagram from the registering vacuum-gauge connected with the smoke-box. The normal draft is shown to be approximately three inches. When the fire-door is opened the draft drops, though the scale at which Fig. 78 is reproduced does not permit this to be well shown. When the ash-pan dampers are closed the draft goes above

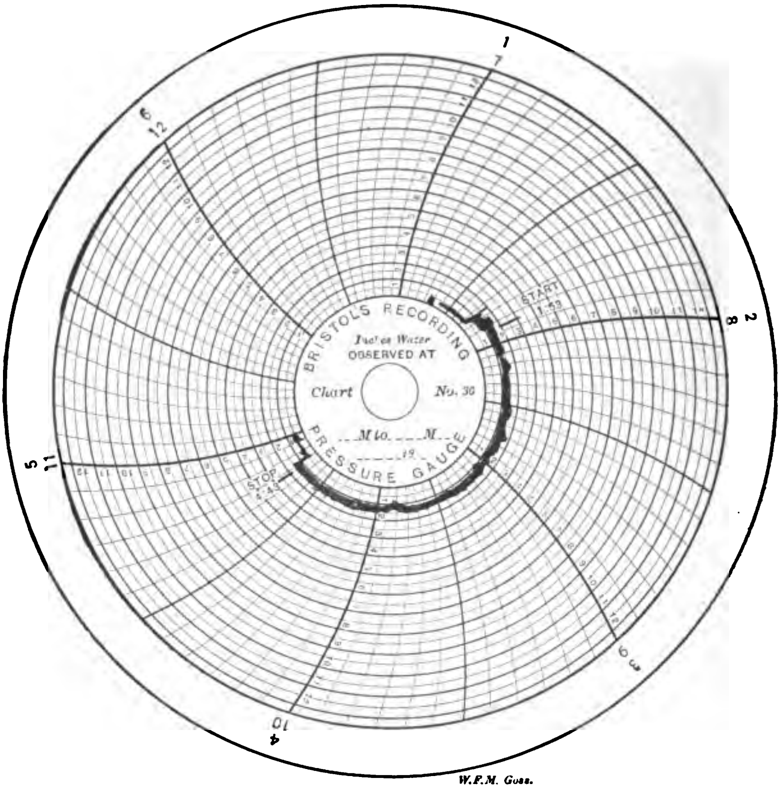

FIG. 79.—Chart from Recording-gauge showing Back Pressure in Exhaust-passage in Saddle.

the normal. The draft readings given (Column 38) are the average results of observations at five-minute intervals. They may be accepted as a close approximation to the normal readings for the tests.

The relation of reduction of pressure in the smoke-box to coal burned per square foot of grate surface is well shown by Fig. 80, and the relation of pressure reduction in smoke-box to evaporation

per square foot of heating surface by Fig. 81. The first diagram (Fig. 80) represents the effect of changes in the draft condition on combustion, and the second (Fig. 81) upon the evaporative power of the boiler. In both diagrams, for reasons already in part explained, the points representing individual tests fall irregularly. An approximation to the mean curve, representing draft and rate of combustion, is shown by the straight line (Fig. 80) which is represented by the equation

$$D=.037G, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which D is the draft in inches of water, and G is the pounds of coal per square foot of grate per hour.

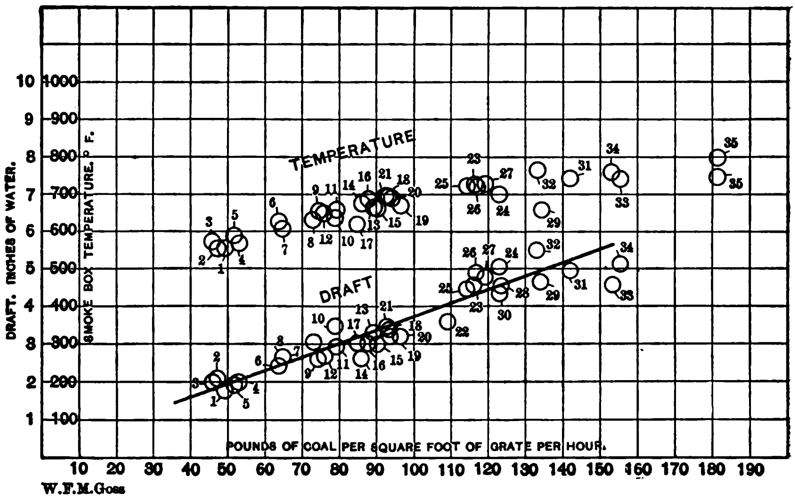

FIG. 80.—Pounds of Coal per Foot of Grate per Hour, as Related to Draft and Smoke-box Temperature.

The smoke-box temperature as affected by changes in the rate of combustion is shown by Fig. 80, and as affected by changes in the rate of evaporation by Fig. 81. From these figures it will be seen that as the power of the boiler is increased the smoke-box temperature rises; also that, as in the case of the draft, the points representing individual tests fall irregularly. It should be noticed, however, that the smoke-box temperature (Column 39) is lower than it is usually assumed to be. It varies from 550 to 798 degrees, a range which, considering the variation in the rate of combustion (Column 40), is not great. Ideal conditions should make the smoke-box temperature a function of the rate of combustion, but under

actual conditions the relationship, as it appears in Fig. 80, is not without variation. This is unquestionably due to differences in fire condition, the efficiency of the action at the grate varying greatly for different tests. Other things being equal, low smoke-box temperature should be expected to indicate a thin fire.

Smoke-box temperatures, plotted with evaporation, are given in Fig. 81. As evaporation is more directly a function of the heat passing the tubes than of furnace action, this comparison does not necessarily involve inequalities in the action of the grate. For this reason the points should be expected to fall more nearly in line, but at the scale chosen for the diagram it must be confessed that the actual difference is not great.

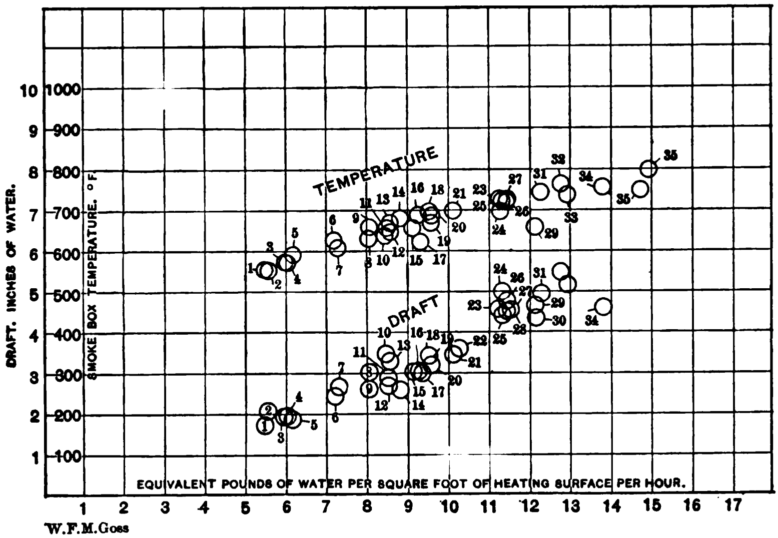

FIG. 81.—Pounds of Water per Foot of Heating Surface per Hour, as Related to Draft and Smoke-box Temperature.

52. Evaporative Performance (Table XXXIII).—If it is remembered that the results of this, as of other tables, are arranged in order of power the data it presents will have increased significance. For example, by merely scanning its columns, the change in evaporative efficiency resulting from increased power may be seen. The table shows the actual evaporation per pound of dry coal (Column 41) to vary from $6\frac{1}{2}$ pounds of water for the lightest power test to 4.77 for the heaviest. Columns 42 and 43 show respectively the evaporation from and at 212 degrees Fahr. for each pound of dry coal, assuming

TABLE XXXIII.

EVAPORATIVE PERFORMANCE.

The several tests represented in this table are arranged in order of the rate of evaporation, No. 1 representing the test for which the rate is least, and No. 35 that for which it is greatest.

Identification of Test.			Evaporative Performance.				
			Evaporation.	Equivalent Evaporation from and at 212° Fahr.			
				Per Pound of Dry Coal.		Per Pound of Combustible.	
Consecutive Number.	Laboratory Designation.	Duration of Test in Minutes.	Total Water Divided by Total Coal.	Assuming Quality of Steam as shown by Column 16.	Assuming all Water Delivered to have been Completely Evaporated into Dry Steam.	Assuming Quality of Steam as shown by Column 16.	Assuming all Water Delivered to have been Completely Evaporated into Dry Steam.
1	2	4	41	42	43	44	45
1	15-1-A	240	6.50	7.83	7.86	8.70	8.74
2	15-1b-V	190	6.90	8.31	8.35	9.24	9.28
3	15-1-H	180	7.67	9.15	9.26	10.17	10.29
4	25-1-A	255	6.70	8.05	8.10	8.95	9.00
5	15-1-G	180	6.98	8.37	8.44	9.31	9.38
6	15-2-A	180	6.63	7.95	8.00	8.84	8.89
7	25-1-V	240	6.61	7.95	7.99	8.84	8.88
8	35-1-A	180	6.44	7.74	7.99	8.61	8.66
9	35-2-F	180	6.31	7.61	7.66	8.45	8.51
10	35-1-V	140	6.26	7.54	7.58	8.38	8.42
11	35-2-E	180	6.26	7.54	7.58	8.38	8.42
12	45-1-A	150	6.62	7.96	7.99	8.83	8.88
13	35-2-B	170	5.62	6.75	6.79	7.50	7.54
14	55-1-A	120	6.00	7.17	7.24	7.95	8.04
15	35-1-H	160	5.95	7.12	7.20	7.92	8.00
16	35-1-G	170	6.19	7.41	7.48	8.24	8.31
17	35-1b-G	170	6.44	7.72	7.80	8.58	8.66
18	25-2-A	180	5.99	7.18	7.23	7.98	8.04
19	55-1-V	120	5.83	6.98	7.03	7.76	7.81
20	35-2b-H	160	5.98	7.18	7.25	7.97	8.05
21	55-1-H	120	6.40	7.68	7.75	8.54	8.62
22	55-1-G	120	5.51	6.59	6.66	7.33	7.40
23	15-9-A	150	5.69	6.81	6.86	7.57	7.63
24	15-9-H	180	5.40	6.47	6.53	7.18	7.26
25	35-2-A	180	5.79	6.94	7.00	7.72	7.78
26	35-3-G	140	5.75	6.87	6.95	7.63	7.73
27	15-9-G	160	5.63	6.74	6.80	7.49	7.56
28	35-3-H	120	5.50	6.59	6.65	7.32	7.39
29	35-2-G	160	5.31	6.35	6.42	7.06	7.13
30	35-2-H	120	5.82	6.95	7.03	7.73	7.81
31	45-2-A	140	5.09	6.09	6.14	6.77	6.83
32	25-3-A	122	5.64	6.76	6.82	7.51	6.57
33	35-2-C	120	4.86	5.86	5.90	6.52	6.55
34	55-2-A	68	5.31	6.34	6.40	7.04	7.11
35	35-3-A	120	4.77	5.71	5.76	6.36	6.39

the quality of the steam to be that shown by the calorimeter (Column 16), and assuming all steam generated to have been dry and saturated. In a similar manner, Columns 44 and 45 show the equivalent evaporation per pound of combustible, assuming the quality of steam to be that shown by Column 16, and assuming the steam to be dry and saturated respectively. All of these values are proportional to the evaporative efficiency of the boiler.

Referring to the equivalent evaporation per pound of coal as it would ordinarily be calculated (Column 42), it will be seen that for the lightest power test the evaporation was 7.83 pounds, and that it diminished greatly, but somewhat irregularly, as the rate of evaporation increased, until, when the power of the boiler became maximum, it was reduced to 5.71, a loss of 27 per cent. The equivalent evaporation per pound of coal for the several tests (Column 42) and the rate of evaporation, as represented by the pounds of water evaporated per square foot of heating surface per hour (Column 21), are

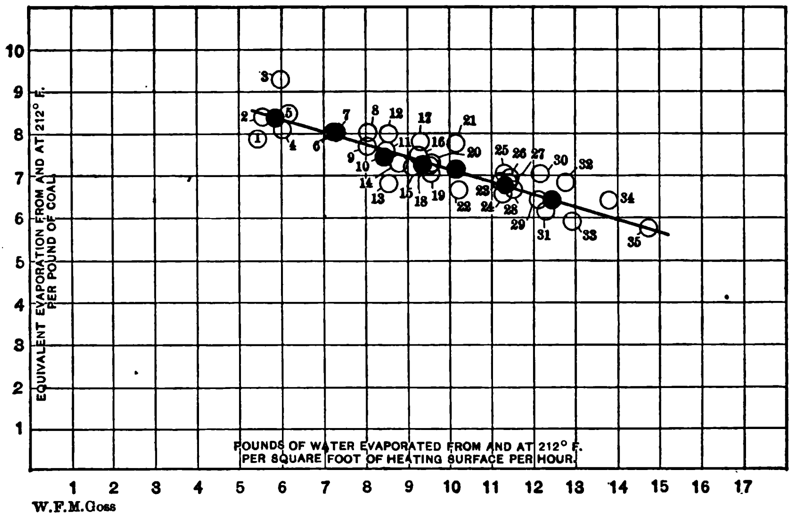

FIG. 82.—Rate of Evaporation and Efficiency, as shown by Pounds of Water per Square Foot of Heating Surface per Hour and Pounds of Water per Pound of Coal.

plotted in Fig. 82. From the points of this diagram an effort has been made to locate a curve which should show the relation of the evaporative efficiency to the rate of evaporation. The method adopted may be described as follows:

The several points were separated into nine groups, those of

each group representing tests of nearly the same power. The grouping is as follows:

1st Group.	2d Group.	3d Group.	4th Group.	5th Group.
Test 1	Test 6	Test 8	Test 15	Test 21
" 2	" 7	" 9	" 16	" 22
" 3		" 10	" 17	
" 4		" 11	" 18	
" 5		" 12	" 19	
		" 13	" 20	
		" 14		

6th Group.	7th Group.	8th Group.	9th Group.
Test 23	Test 29	Test 34	Test 35
" 24	" 30		
" 25	" 31		
" 26	" 32		
" 27	" 33		
" 28			

The centers of the first seven groups have been determined and their location is shown on the diagram by solid black spots. A straight line drawn as nearly as possible through the points thus located is assumed to show the relationship sought. There are but two centers of groups that are as much as one per cent away from this line, and four touch it within one-tenth of one per cent.

Before attempting a more critical examination of the line thus located, it may be well to inquire why so many of the points representing individual tests, and shown upon the diagram by light circles, fall at so great a distance from it. The facts in the case are as follows:

11	points or	31	per cent.	of the	whole	agree	with the	curve	within	1	per cent.
15	"	43	"	"	"	"	"	"	"	2	"
18	"	51	"	"	"	"	"	"	"	3	"
21	"	60	"	"	"	"	"	"	"	4	"
24	"	69	"	"	"	"	"	"	"	5	"
27	"	77	"	"	"	"	"	"	"	6	"

There is a wide-spread feeling among motive-power men that the character of the exhaust has much to do with the efficiency of the furnace action; that a heavy exhaust incident to slow running will have a different effect upon the fire than the lighter but more rapid action attending higher speeds, though the draft, as registered by a gauge in the smoke-box, may read the same. It was but natural, therefore, to first look to the engine conditions for an explanation of the irregularities in the efficiency of the boiler. The result of such a study tends to disprove the commonly accepted theory, and justifies

the conclusion that, with a given vacuum in the smoke-box, the action of the boiler is quite independent of the manner in which the vacuum is maintained, whether by slow heavy beats or quicker, lighter pulsations. For example, we have tests 1, 2, 3, and 5, for which differences in engine conditions were confined to the dimensions or the setting of the valves. All were run at a speed of 15 miles an hour, and all at shortest cut-off, and yet the results are widely separated. In the second group are tests 6 and 7, which check each other closely, but one was run at low speed under a liberal cut-off (15–2), while the other was at a higher speed and shorter cut-off (25–1). The third group consists of tests 8 (35–1–A), 9 (35–2–F), 10 (35–1–V), 11 (35–2–E), 12 (45–1–A), 13 (35–2–B), and 14 (55–1–A). Of tests numbered 8, 9, 10, and 11, all at 35 miles an hour, two were run at short cut-off, and two at a more liberal one, but the points of all fall near the curve. Of the fourth group, including tests 15 (35–1–H), 16 (35–1–G), 17, (35–1_b–G), 18 (25–2–A), 19 (55–1–V), 20 (35–2_b–H), tests 16, 18, and 20, which differ one from the other in speed and cut-off, fall most nearly upon the curve, while tests 15 and 16, having the same conditions, fall on the opposite side. The two tests of the fifth group, 21 (55–1–H) and 22 (55–1–G), both at the same high speed and short cut-off, fall on either side of the line curve. A comparison of the remaining groups gives similar results. It seems, therefore, but fair to conclude that variations in the character of the exhaust-jet—such as result from changes in speed or cut-off—do not in themselves affect the efficiency of the boiler.

It is well known that when an engine is working under a light load, a skillful fireman can maintain a very thin fire over the whole grate. Under favorable conditions one may almost see the grate-bars through the fire. All portions of such a fire are bright, and there will be plenty of steam, but the firing must be frequent. Such a fire offers so little resistance to the incoming air that larger volumes than are needed for combustion pass through the furnace and absorb a portion of its heat. Again, the fire may be made so thick that it will not burn clear. The demand for steam may not be great, and a sluggish and smoky fire may serve to generate it. But such conditions cannot give maximum efficiency. It follows that somewhere between the very thin fire and the very thick fire will be one of such thickness as will result in maximum efficiency. For all tests here reported the same fireman served. He was skilled in his work, and every effort was made to have a fire suited to the demands

made upon it. But the locomotive fireman takes his cue from the steam-gauge rather than from the furnace. If the finger of the gauge is moving upward, or holds its own, the fire is usually assumed to be all right; if it falls, something must be done. The actual condition of the fire under such circumstances, especially in a test for which a constant speed and load are maintained, depends very much upon the character of the fire at the start. It may happen that two tests, apparently identical as to speed, load, etc., may be run, one with a thick fire and the other with a comparatively thin fire, and, so far as outward conditions are concerned, the tests may seem equally satisfactory, while neither satisfies conditions for maximum efficiency.

With these facts in mind, we may now inquire further concerning the variations in the results of tests to which attention has already been directed. Thus, a comparison of the 11 tests that agree with the mean curve (Fig. 82) within 1 per cent with the 11 tests which have the greatest divergence from it reveals the fact that the tests of the latter class are, for the most part, those in which the firing required unusual care. The greater part of the tests of this group are either tests at very low power, for which only a light fire could be maintained without danger of losing steam at the safety-valve, or tests at high speeds, when the work of firing was hard and difficult. Included in this group also is one test under low boiler pressures, which is to be regarded as a light power test. On the other hand, those which agree most nearly with the curve are, for the most part, tests at medium load, which were easily fired. An exhibit of these facts is as follows:

Results in Agreement with the Curve.			Results which Diverge from the Curve.		
Distance which Points are off the Curve. Per Cent.	Test Number.	Laboratory Designation. Speed. Cut-off. Series.	Distance which Points are off the Curve. Per Cent.	Test Number.	Laboratory Designation. Speed. Cut-off. Series.
0	6	15–2–A	10.6	13	35–2–B
0.2	35	35–3–A	9.9	3	15–1–H
0.3	11	35–2–E	8.5	21	55–1–H
0.4	7	25–1–V	7.9	1	15–1–A
0.4	10	35–1–V	7.4	32	25–3–A
0.5	8	35–1–A	7.3	30	35–2–H
0.7	23	15–9–A	6.5	22	55–1–G
0.8	27	15–9–G	6.2	33	35–2–C
0.9	20	35–2_b–H	5.8	34	55–2–A
0.9	16	35–1–G	5.4	31	45–2–A
0.9	2	15–1_b–V	5.3	12	45–1–A

Accepting the experimental results as reliable, it seems safe to conclude that variations in the efficiency of the boiler, as disclosed by different tests at the same power, are due to irregularities in the character of firing.

53. Power and Efficiency.—Referring again to Fig. 82, it should be noted that the ordinates in this diagram represent the evaporative efficiency of the boiler, and the abscissæ the rate of evaporation. The manner in which the line which is assumed to represent the mean of these points was drawn has already been described. The equation for the line is

$$E = 10.08 - .296H, \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

in which E is the pounds of water evaporated from and at 212 degrees Fahr. per pound of coal, and H the pounds evaporated per square foot of heating surface per hour.

This equation and others derived from it are assumed to represent the average performance of the boiler when using Indiana block coal. By its use it is possible to obtain a coal record from the water rate, no weighing being made of the fuel. Defense for such a practice is to be found in the comparative ease with which the water record is obtained, and in the fact that the coal consumption, as determined from the equation, is a more consistent factor than can ordinarily be obtained experimentally from a few tests. The form of the equation will doubtless hold for all boilers of similar design with that tested, but the constants may change with the proportions of the boiler, and will of necessity change with the character of the fuel employed.

54. Efficiency as Affected by the Quality of Fuel.—While apparently somewhat apart from the present purpose, it will be of interest, in connection with the general discussion, to review certain results which have been obtained from five different samples of fuel tested in the same boiler, and which were reported in a paper presented to the Western Railway Club at the December meeting, 1898. The several samples were designated as A, B, C, D, and E. All were bituminous coals. The evaporation obtained from each of these samples is shown by Fig. 83. Line D on the diagram very nearly corresponds with that given in Fig. 82 for the Indiana block, and its equation is substantially that given above. The equation for line E, representing the best coal, may be taken as

$$E_1 = 12.9 - 0.41H, \quad . \quad . \quad . \quad . \quad . \quad . \quad (a)$$

and for line C, representing the poorest coal, as

$$E_2 = 9.4 - 0.24H. \quad \ldots \ldots \ldots \quad (b)$$

These equations probably represent the range of variations in performance as affected by different qualities of fuel.

The rate of evaporation represented by the experiments with the Indiana block, upon which equation 4 is based, lies between the limits of 5 and 15 pounds of water per square foot of heating surface per hour, while the limits on which equations a and b are based are but little narrower. The equations, then, are reliable when H is allowed a value which is not less than 5 nor greater than 15.

In this connection it is of interest to note that the lines E, A, B, C, D converge, and it may be assumed that if D and C were sufficiently extended they would meet; that is, if the rate of evaporation were made sufficiently high, both the good and the poor coal would give the same evaporation. The point where this would happen can be determined from equations a and b, by making E_1 equal to E_2. Thus,

$$12.9 - 0.41H = 9.4 - 0.24H$$

and

$$H = 20. \text{ approximately.}$$

That is, accepting for the moment this equation as true for all values of H, they show that when the boiler is forced to evaporate 20 pounds of water per square foot of heating surface per hour, the poorer coal will evaporate as much water per pound as the better. It is evident that the equations are not to be relied upon for conditions so widely separated from those covered by the experiments, and it is equally evident that the boiler could not easily be forced to so high a rate of evaporation. The general conclusion to be deduced is, however, perfectly logical. The higher the power to which a boiler is forced, the smaller is the fraction of the total heat developed which can be absorbed by the heating surface. If forced to very high power, the amount of heat utilized out of all that is available becomes so small that slight variations in the amount available do not measurably affect the amount utilized.

If carried to extreme limits, it will doubtless appear that the line of Fig. 82, represented by equation 4, is in fact not straight, though, within limits which are sufficiently broad to cover all practical cases,

it may probably be so considered. The form of this and of other similar lines is the subject of discussion in a preceding paragraph.

55. Derived Relations.—The relation between the rate of evaporation and the rate of combustion for the thirty-five tests under discussion is shown by Fig. 84. The points in this figure are located from experimental data, but the curve which is assumed to rep-

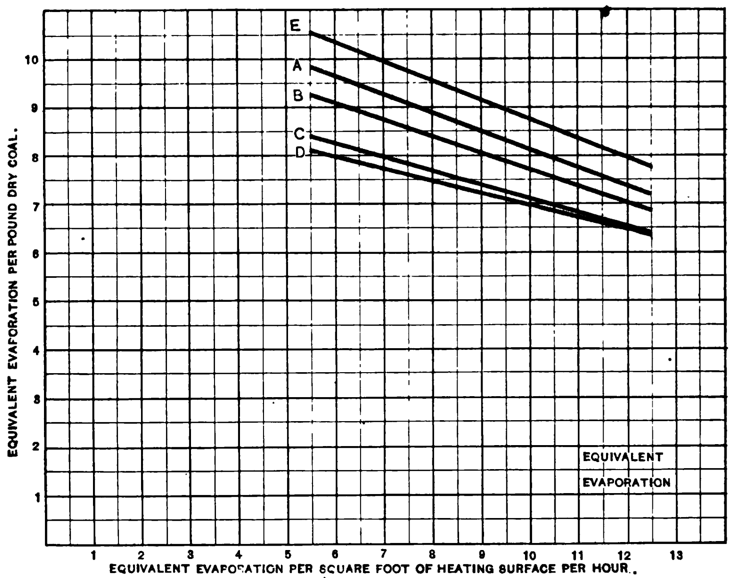

FIG. 83.—Rate of Evaporation and Efficiency for Five Different Samples of Bituminous Coal.

resent their mean value was plotted from an equation derived from equation 4. Thus equation 4, as already given, is

$$E = 10.08 - .296H. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

We may let W = total pounds of water evaporated, from and at 212 degrees Fahr. per hour, and

$$C = \text{total pounds of coal fired per hour;}$$

We may note also that 1,214.4 = square feet of heating surface in the experimental boiler. Then

$$\frac{W}{C} = E, \quad \text{and} \quad H = \frac{W}{1{,}214};$$

or

$$\frac{W}{C} = 10.08 - 0.296H$$

$$= 10.08 - \frac{0.266W}{1,214.4},$$

and

$$C = \frac{W}{10.08 - 0.000244W}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

It is from this equation that the curve (Fig. 84) was plotted. It will be seen that, while the curve is derived quite independently of the points, the two systems agree closely.

The relation between water evaporated per pound of coal and pounds of coal consumed per square foot of grate per hour is shown by Fig. 85.

This curve, in common with the one preceding, is plotted from its equation, which was obtained as follows:

Having $$E = 10.08 - .296H, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

let G = pounds of coal per square foot of grate per hour. Note that 17.25 = square feet of grate surface for the experimental boiler. It has already been shown that

$$\frac{W}{C} = 10.08 - \frac{0.296}{1,214.4}W,$$

or

$$\frac{W}{C} + .000244\frac{W}{C}C = 10.08;$$

but

$$\frac{W}{C} = E,$$

therefore

$$E + 0.000244EC = 10.08,$$

and

$$E = \frac{10.08}{1 + 0.000244C}.$$

But

$$C=17.25G,$$

and therefore

$$E=\frac{10.08}{1+.00421G}. \quad \quad (6)$$

The agreement between the curve plotted from this equation and the experimental points is as close in this case (Fig. 85) as in those previously discussed. Other relationships may be established by aid of those already given. Perhaps the most interesting is that of draft (D) to total weight of water evaporated per hour (W), which takes the form of

$$D=\frac{.00214W}{10.08-.000244W}. \quad \quad (7)$$

Before leaving this phase of the subject, it is of interest to note that the plotted curves representing the derived equations (Figs. 84

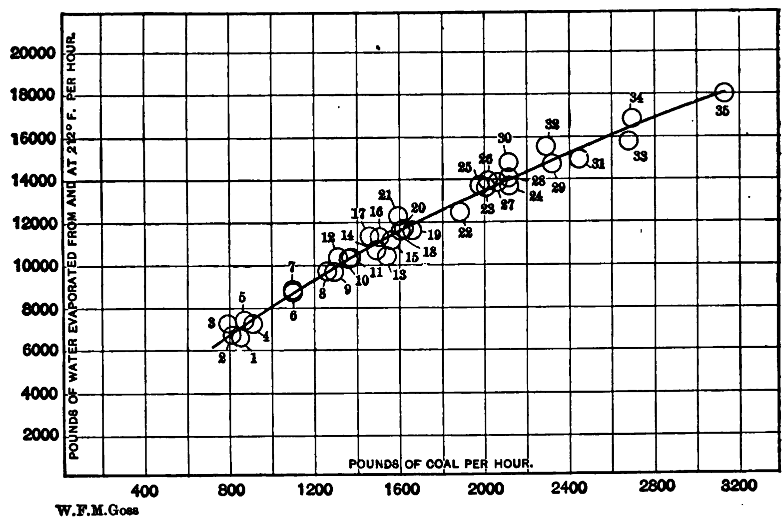

FIG. 84.—Combustion and Evaporation, as shown by Total Pounds of Coal per Hour and Total Pounds of Water per Hour.

and 85) are based upon the equation of a straight line. This straight line (Fig. 82) is believed to fairly represent the experimental points for which it stands, and it follows that the curves of Figs. 84 and 85 represent, with an equal degree of accuracy, the experimental points

in the midst of which they are drawn. If, however, the experimental points represented by Fig. 85 had been accepted as a starting-point for the several curves, a straight line might have been drawn through them without difficulty. Had this been done and the relation represented by Fig. 82 been mathematically derived from it, the line of Fig. 82 would have been a curve. From these considerations, and from those previously presented concerning the convergence of the several lines representing different samples of coal, it is evident that none of the relationships discussed are perfectly represented by a straight line. But with nothing but the experimental point as a basis,

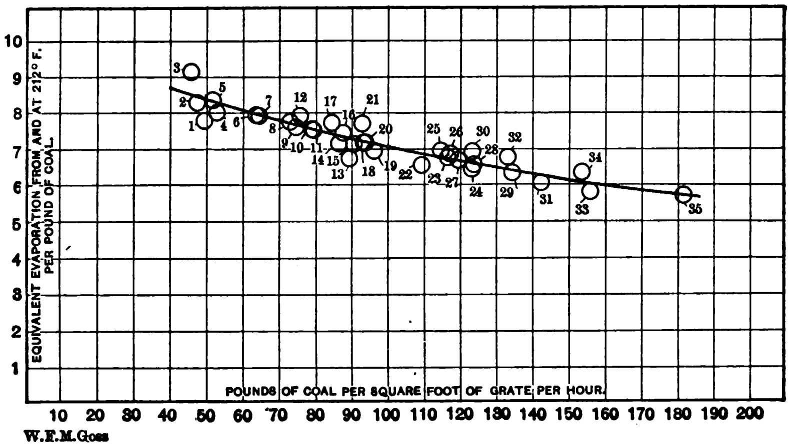

FIG. 85.—Rate of Combustion and Efficiency, as shown by Pounds of Coal per Square Foot of Grate per Hour and Pounds of Water per Pound of Coal.

it appears difficult to locate a line which will better represent them than the straight line of Fig. 82, and as before noted, within the limits for which it applies, such a line cannot be much in error.

56. Conclusions.—1. The steam delivered by the boiler tested under constant conditions of running, as shown by a calorimeter attached to the dome, is at all times nearly dry, the entrained moisture rarely equaling 1.5 per cent, and being generally much less than this. While the relationship cannot be perfectly defined, it appears that the entrained moisture increases slightly as the rate of evaporation is increased.

2. The maximum power at which the boiler was worked with Brazil block coal was such as gave 30 boiler horse-power for each square foot of grate, and .427 horse-power for each square foot of heating surface. Experiments with other fuels indicate that

these values may be increased by the use of a better coal by about 15 per cent, giving maximum values, which, in round numbers, are 35 horse-power per square foot of grate and .5 horse-power per square foot of heating surface. For the type of boiler experimented upon, and under conditions of constant running, these values may be accepted as near the maximum.

3. The maximum rate of combustion reached was 182 pounds of coal per square foot of grate per hour, which is equivalent to 2.6 pounds per square foot of heating surface.

4. The maximum draft for any test was that for which the average value was 7.5 inches. If D is the reduction of pressure in the smoke-box measured in inches of water, and G the pounds of coal burned per square foot of grate per hour, then

$$D=.037G.$$

Also, if W be the total weight of water evaporated per hour, the draft necessary to produce a given evaporation is represented by the equation

$$D=\frac{.00214W}{10.08-.000244W}.$$

These equations apply to the boiler tested when using Indiana block coal.

5. Smoke-box temperature ranges from 550 to 800 degrees Fahr., values which are lower than those which are often assumed to prevail.

6. The evaporative efficiency of the boiler as affected by different rates of evaporation is expressed by the equation

$$E=10.08-.296H,$$

in which E is the pounds of water evaporated from and at 212 degrees Fahr. per pound of coal, and H the pounds of water evaporated from and at 212 degrees Fahr. per square foot of heating surface per hour; this for the boiler tested using Indiana block coal, and for values of H of not less than 5 nor greater than 15. With different coals the constants will vary, results which are near the minimum being expressed by

$$E_{\text{min.}}=9.4-.024H,$$

and results near the maximum by

$$E_{\text{max.}} = 12.9 - .041H.$$

These equations may be accepted as of rather general application, representing approximately the performance of any locomotive boiler.

7. The evaporative efficiency of the boiler as affected by different rates of combustion is expressed by the equation

$$E = \frac{10.08}{1 + .00421G},$$

in which E, as before, is the pounds of water evaporated from and at 212 degrees Fahr. per pound of coal, and G the pounds of coal burned per square foot of grate per hour; this for the boiler tested using Indiana block coal.

8. The relation of coal burned to water evaporated is expressed by the equation

$$C = \frac{W}{10.08 - .000244\,W},$$

in which C is the total pounds of coal burned per hour, and W the total pounds of water evaporated from and at 212 degrees Fahr. per hour; this for the boiler tested using Indiana block coal.

9. The condition of running the engines, whether with long or short cut-off, or at high or low speed, does not appear to affect the efficiency of the boiler of a locomotive, except in so far as it affects the average value of the draft.

10. The efficiency of the boiler of a locomotive, as disclosed by two different tests, for which all conditions of running are the same, may vary considerably, due doubtless to inequalities in firing.

CHAPTER VII.

HIGH RATES OF COMBUSTION AND BOILER EFFICIENCY.*

57. General Statement.—The fact has already been established that, within limits defined by practice, the boiler of any given locomotive is most efficient when worked at its lowest power (Chapter VI.). The power of such a boiler depends upon the rate at which coal is fed to the furnace, but the return, in water evaporated for each pound of coal burned, is reduced as the rate of combustion is increased.

It has been found that for any given boiler there is a definite relationship between the evaporation per pound of coal and the weight of coal fired per hour, and for the boiler of the Purdue locomotive this relationship has been determined. (See equation 6 and Fig. 85, Chapter VI.) The facts show that when coal is burned at the rate of 50 pounds per square foot of grate per hour, 8.3 pounds of water are evaporated for each pound of coal; while if the rate of combustion is increased to 180 pounds per foot of grate, the evaporation falls to 5.7 pounds—a loss in water evaporated per pound of coal of about 30 per cent. This loss may be due to a failure of the heating surfaces to absorb properly the increased volume of heat passing over them, or to the imperfect combustion of the fuel upon the grate, or it may be due to a combination of these causes.

That a portion of the loss occurs along the heating surfaces hardly admits of question, since it is well known that any increase in the rate of combustion results in a rise in the temperature of the smoke-box gases; but whether, under ordinary conditions, any considerable portion of the loss accompanying increased rates of power is due to

* The experiments discussed in this chapter were presented in a paper before the New York Railway Club at a meeting held in September, 1896, entitled "Effects of High Rates of Combustion upon the Efficiency of Locomotive Boilers." The experimental facts which are here given are unchanged, but the discussion of them is somewhat altered to include facts of more recent development, and the conclusions are considerably modified.

imperfect combustion had not been demonstrated previous to the publication of the results herein discussed, and it is this question especially that the present chapter attempts to treat.

It will be seen that a separation of the losses which may occur at the grate from those which take place along the heating surface could not be accomplished by boiler-tests alone, because the results of such tests give the combined effect of both these losses. There are two variables involved, and in order that either may be determined one must be given a constant value. In the tests described, action along the heating surface was maintained constant, while conditions at the grate were varied.

Tests were outlined in which the total weight of fuel fired was to be constant throughout the series, while the rate of combustion was to be made different for each test by changing the area of the grate. It is evident that if the action at the grate were equally efficient during the several tests—that is, for different rates of combustion—this provision would cause the same amount of heat to pass over the heating surfaces of the boiler, and hence would produce the same evaporation and the same smoke-box temperature. If, on the other hand, the combustion should prove less efficient for any one test than for others, a smaller quantity of heat would sweep the heating surface, less water would be evaporated, and the smoke-box temperature would probably be lower.

The outline provided for all observations usual in boiler-testing, and, in addition to these, for a determination of the weight of fuel lost in the form of sparks, and for chemical analyses of the fuel used, of the sparks caught, and of the smoke-box gases.

58. The Tests and the Results.—The first test was run with the locomotive under normal conditions. The whole grate was covered with fuel, the throttle was fully open, the cut-off approximately 6 inches, and the load such as to make the speed 25 miles per hour. These conditions gave a rate of combustion of 61 pounds of coal per square foot of grate per hour.

In preparation for the second test, one-quarter of the grate was made non-effective, or "deadened," by a covering of fire-brick (Fig. 86). The exhaust-tip was reduced, so that while the engine was running as before and using approximately the same amount of steam, the same total weight of fuel could be burned on the reduced grate as in the first test had been burned on the whole grate. Trial tests were run, until it was known that the changes made would permit the

desired conditions to be maintained. The rate of combustion in this test was 84 pounds per square foot of grate area.

In preparation for the third test, the grate surface was reduced

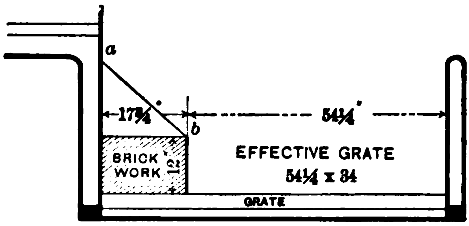

FIG. 86.—Test 2.

to half its original area (Fig. 87), and the rate of combustion was increased to 124 pounds per square foot of grate area; and during

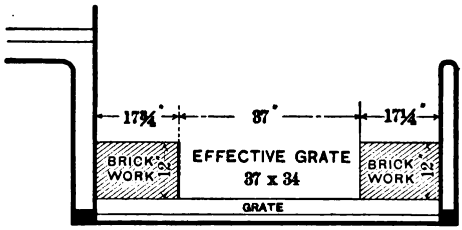

FIG. 87.—Test 3.

the fourth test only one-quarter of the original grate was used (Fig. 88), the combustion in this case rising to 241 pounds per square foot of grate surface.

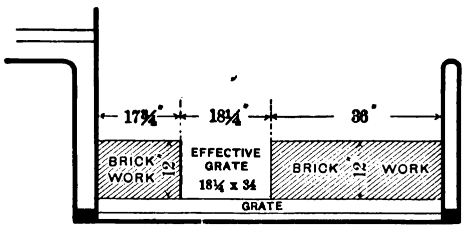

FIG. 88.—Test 4.

It should be evident, from what precedes, that the prescribed conditions were designed to make each test a duplicate of every other test, excepting in the matter of grate area, this being the one variable for the series.

The coal used in the several tests was of uniform quality, the chemical analyses showing no greater variation than might occur in different samples from a single shipment. The maximum weight of coal fired per hour in any test was 1,087 and the minimum was 1,038, a difference of less than 50 pounds in more than a thousand, while the variation during three of the four tests does not exceed 1.2 per cent of the weight fired. All firing was done by one man, the attendants engaged in taking the more important observations were the same for all tests, and all external conditions affecting the action of the boiler were uniform throughout the series.

Table XXXIV contains the complete records of the tests.

59. Grate Losses.—Since in the series under consideration, the volume of heat sweeping the heating surface was the same for all tests, it is reasonable to conclude that any loss in evaporative efficiency attending increased rates of combustion is due to the action which goes on at the grate. The extent of such loss appears in Table XXXV, Item 3 of which gives the evaporation per pound of coal for each test, and Item 4 the loss in evaporation in terms of that of Test 1. The results show that with each increase in the rate of combustion, the evaporative efficiency diminishes. When the combustion is forced to 241 pounds of coal per foot of grate surface per hour, the loss in the evaporation per pound of coal, in terms of that which was obtained when the rate of combustion was 61 pounds, is 19 per cent. This loss at the grate may have its source in any one or all of several causes: (1) A portion of the coal delivered to the fire-box may be dissipated in the form of sparks; (2) the oxidation of the combustible gases may be incomplete; (3) the fire-box may be cooled by excess air.

The probable extent of losses resulting from these causes will next be considered.

60. Spark Losses as a Factor Affecting Grate Losses.—The data show that a large portion of the loss which occurs at the grate when the rate of combustion is increased, is due to losses of sparks. The facts involved are well shown by Table XXXVI. A comparison of values under Item 6 of this table will show the rate of increase in the spark losses with increased rate of combustion. For example, when the rate of combustion is 61 pounds, 4.3 per cent of the coal fired is accounted for as sparks, whereas when the rate of combustion is increased to 241 pounds, 15 per cent of the coal fired is thus lost. The rate of increase in spark production may best be seen by assuming

TABLE XXXIV.

OBSERVED AND CALCULATED DATA.

1. Test number	1	2	3	4
2. Month and day (1896)	Feb. 8	Feb. 11	Feb. 15	Feb. 22
3. Duration of test, hours	6	6	6	6
4. Approximate portion of whole grate used	Full	Three-fourths	Half	One-fourth
5. Exact area of effective grate, square feet	17.50	13.01	8.74	4.31
6. Barometric pressure, pounds	14.41	14.43	14.34	14.47
*Analysis of Coal.**				
7. Per cent fixed carbon	49.65	51.84	51.09	51.59
8. Per cent volatile matter	40.29	39.00	38.93	38.87
9. Per cent combined moisture	3.15	3.62	2.35	3.44
10. Per cent ash	6.91	5.54	7.63	6.10
Coal (Brazil Block).				
11. Pounds fired	6522	6628	6716	6328
12. Weight of water in each pound of coal fired	0.012	0.016	0.030	0.012
13. Pounds of dry coal for test	6443	6522	6514	6227
14. Pounds of dry coal per hour	1074	1087	1086	1038
15. Pounds of dry coal per hour per square foot of grate	61.4	83.5	124.2	240.8
16. Pounds of combustible for test	5792	5921	5856	5635
17. Percentage of fixed carbon in coal, dry and free from ash	56	57	57	57
18. Approximate number of B.T.U. per pound of combustible	13800	14040	14040	14040
19. Approximate number of B.T.U. per pound of dry coal	13000	13000	13000	13000
20. Theoretical evaporation from and at 212° per pound of dry coal	13.46	13.46	13.46	13.46
Ash.				
21. Pounds of dry ash in ash-pan for test	446	396	297	164
22. Pounds of ash in coal fired as shown by analysis of coal	445	361	497	380
23. Pounds of ash by analysis, minus pounds found in ash-pan	−1	−35	200	216
*Analysis of Sparks.**				
24. Per cent of fixed carbon	61.74	64.88	71.32	76.44
25. Per cent volatile matter	4.36	4.16	3.45	3.29
26. Per cent combined moisture	1.82	1.82	1.66	1.86
27. Per cent ash	32.08	29.14	23.57	18.41
Sparks.				
28. Pounds caught in front end during test	75	213	494	566
29. Pounds passing out of stack for test	294	358	278	492

OBSERVED AND CALCULATED DATA—(*Continued*).

Test number	1	2	3	4
30. Total pounds of sparks for test	369	571	772	1058
31. Pounds of sparks per square foot of grate per hour	3.5	7.3	14.7	41.0
32. Pounds of combustible in sparks for test	242	395	576	837
33. Percentage of fixed carbon in sparks dry and free from ash	94	94	95	96
34. Approximate B.T.U. per pound of sparks	9870	10360	11200	11880
35. Pounds of coal equivalent in heating value to one pound of sparks	0.75	0.80	0.86	0.91
36. Pounds of coal equivalent in heating value to total weight of sparks for test	277	457	664	963
*Analysis of Smoke-box Gases.**				
37. Per cent carbon dioxide	5.45	6.25	4.80	1.80
38. Per cent heavy hydrocarbons	0.50	0.40	0.40	0.50
39. Per cent oxygen	12.15	11.80	14.60	18.70
40. Per cent carbon monoxide	0.00	0.00	0.00	0.55
41. Per cent nitrogen	81.90	81.55	80.20	78.45
Other Smoke-box Data.				
42. Diameter of double exhaust-tip, inches	3	2.75	2.35	1.75
43. Draft in inches of water	2.2	2.5	3.3	5.6
44. Temperature of smoke-box, deg. F.	647	629	610	500
Water and Steam.				
45. Pounds of water delivered to boiler	44756	43081	40710	34770
46. Temperature of feed, deg. F.	54.0	53.0	53.0	52.7
47. Boiler pressure, by gauge	129.4	127.2	127.2	129.1
48. Quality of steam in dome	0.982	0.981	0.984	0.983
Evaporation.				
49. Pounds of water evaporated per pound of dry coal	6.94	6.60	6.30	5.58
50. Equivalent evaporation from and at 212° F.	8.26	7.87	7.52	6.67
Horse-power.				
51. Horse-power of boiler	257	248	226	201
52. Horse-power per square foot of grate	15	19	26	47
Approximate Efficiency.†				
53. Ratio of heat developed in the furnace to heat absorbed by water	0.61	0.59	0.56	0.50

* All chemical analyses were made, under the direction of Professor W. E. Stone, by Charles D. Test, A.C.

† The efficiency is approximate only, since the heating value of the coal is only approximately known. But as the same coal was used for all tests, there can be no error in using his factors for purposes of comparison within the limits of the present series of tests.

the spark loss for Test No. 1 to be zero, the values for all tests then becoming 4.3 per cent less than those given in Item 6. The increase on this basis is shown by Item 7.

TABLE XXXV.

1. Number of test	1	2	3	4
2. Rate of combustion, pounds of coal per foot of grate surface per hour	61	84	124	241
3. Equivalent evaporation from and at 212° F. per pound of coal, pounds	8.26	7.87	7.52	6.67
4. Loss of evaporation in terms of the evaporation for Test No. 1, per cent	0.0	4.7	9.0	19.2

By comparing Item 4, of Table XXXV., with Item 7, of Table XXXVI., the proportion of the whole loss occurring at the grate, which is accounted for as sparks, may be made to appear. Such a comparison is presented as Table XXXVII. In this table, Item 3 presents the loss by evaporative efficiency as the rate of combustion is increased, Item 4 the fuel loss due to sparks, and Item 5 the difference. The significance of this comparison will appear when it is remembered that if spark production accounted for all of the loss which has been shown to occur at the grate, the values of Item 4 would be equal to those of Item 3. The difference, as disclosed by Item 5, is therefore the loss unaccounted for which occurs at the grate. It is relatively very small, representing only 8 per cent of the fuel fired when the abnormally high rate of combustion of 240 pounds is maintained.

TABLE XXXVI.

1. Number of test	1	2	3	4
2. Rate of combustion, pounds of coal per foot of grate surface per hour	61	84	124	241
3. Total pounds of coal per hour	1,074	1,078	1,086	1,038
4. Total pounds of sparks per hour	61.5	95.1	128.6	176.3
5. Fuel value of sparks, B.T.U. per pound	9,870	10,360	11,200	11,880
6. Value of spark losses in per cent of coal fired	4.3	7.2	10.2	15.5
7. Increase of spark losses over those of Test No. 1, in per cent of coal fired	0.0	3.9	5.9	11.2

TABLE XXXVII.

1. Number of test	1	2	3	4
2. Rate of combustion, pounds	61	84	124	241
3. Loss of evaporation in terms of the evaporation for Test No. 1, per cent	0.0	4.7	9.0	19.2
4. Fuel loss by sparks in excess of those lost in Test No. 1, per cent	0.0	3.9	5.9	11.2
5. Difference unaccounted for (Item 3–Item 4)	0.0	0.8	3.1	8.0

61. Losses Due to Incomplete Combustion and Excess Air.—The tests show losses on these accounts to be small, since all loss occurring at the grate not accounted for as sparks is that appearing in Item 5 of Table XXXVII. In service the rate of combustion does not often exceed 125 pounds of coal per hour, for which all losses unaccounted for, which must include incomplete combustion and excess air, do not exceed 3 per cent.

62. Losses Along the Heating Surface.—In the special tests under consideration the heating surfaces were swept by the same volume of heat throughout the series. By comparing the evaporation per pound of coal under different rates of combustion for these special tests with that evaporation which results under normal conditions of operation, there will be shown the extent of the loss which under normal conditions occurs along the heating surface when the rate of combustion is increased. The facts involved by such a comparison are set forth in Table XXXVIII. Item 3 of this table gives the evaporation per pound of coal as derived from the series of special tests for which different rates of combustion were employed while maintaining uniform action along the heating surface. These values show, as already explained, those losses only which occur at the grate. Item 4 gives the evaporation per pound of coal under normal conditions of operation, as obtained from the equation deduced from thirty-five tests as explained in Chapter VI. A comparison of the values of this item discloses all losses which under normal conditions occur with increased rates of combustion; they include both those losses which appear in Item 3 and also those which occur along the heating surface. The loss which under normal conditions occurs along the heating surface when the rate of combustion is increased should, therefore, appear in the difference between the values of Items 3 and 4. This is given as Item 5. To make such

a comparison effective, however, there is need of one correction. Thus, values for Test 1, Items 3 and 4, are assumed to represent identical conditions and should in fact be the same. The difference of .26 of a pound is due to the fact that the experimental result (Item 3) does not fall quite on the curve based upon many different tests from which the equation underlying Item 4 was derived. The facts are best shown by Fig. 89, in which the dotted line represents the

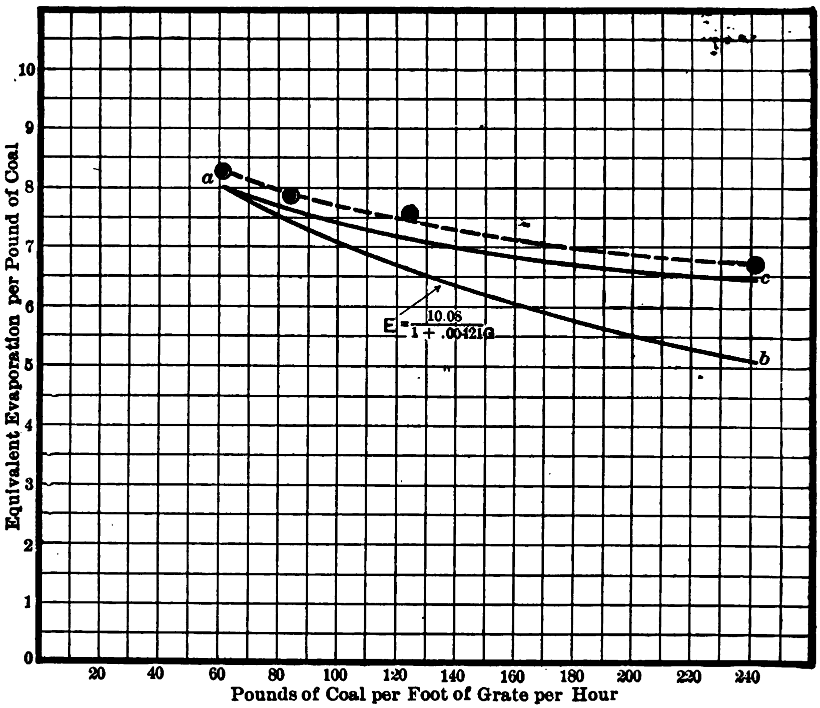

FIG. 89.—Evaporative Efficiency and Rate of Combustion.

special tests, and the curve *ab* the standard equation for the experimental boiler. Since the present discussion is chiefly concerned in the relative slope of the curves, comparison will be facilitated if they have one common point. This can be accomplished by dropping the curve of the special tests down to the position *ac*, Fig. 89, which involves a reduction of the several values of Item 3 (Table XXXVIII). The corrected differences then become those appearing as Item 6. These values are comparable with those of the last items of the three tables immediately preceding.

TABLE XXXVIII.

1. Number of test	1	2	3	4
2. Rate of combustion, pounds of coal per foot of grate surface per hour	62	84	124	241
3. Equivalent evaporation from and at 212° F. per pound of coal fired in special tests	8.26	7.87	7.52	6.67
4. Equivalent evaporation, from and at 212° F. per pound of coal fired, which would have been obtained had the same rate of combustion been maintained over the whole area of the grate: $E=\frac{10.08}{1+.00421G}$	8.00	7.38	6.63	5.01
5. Difference (3–4)	0.26	0.49	0.89	1.66
6. Difference, Item 5 less .26	0.00	0.23	0.63	1.40

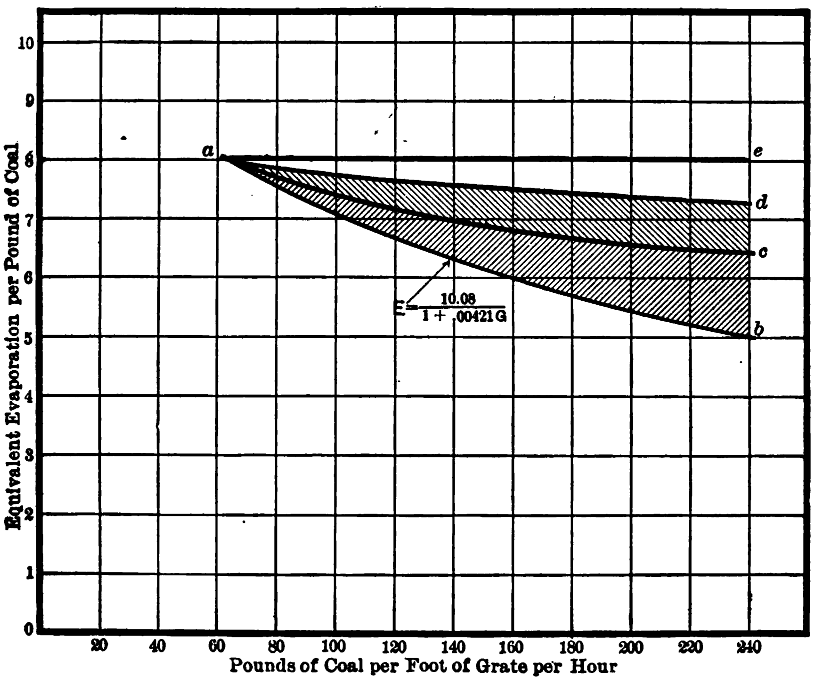

FIG. 90.—Losses in Evaporative Efficiency.

63. Conclusions.—The significance of the results disclosed by Tables XXXV. to XXXVIII., inclusive, are well shown by means of Fig. 90, in which the curve *ab* represents the normal evaporative

efficiency of the boiler under different rates of combustion. The curve *ac* represents the evaporative efficiency under the special tests for which losses along the heating surfaces are eliminated. The curve *ad* represents the results which would have been obtained from the special tests had there been no spark losses, and the line *ae* represents an evaporative efficiency of constant value. Under normal conditions of operation, the area *bac* represents that portion of the loss which takes place along the heating surface. The area *cae* represents that portion of the loss which takes place at the grate, of which loss that represented by the area *cad* is known to be due to spark production, while that represented by the area *dae* remains unaccounted for.

Speaking in very general terms, it appears that, under normal conditions of service, about one-half of the heat loss which results from forcing a boiler to higher power takes place along the heating surface and is, of course, unavoidable. Of the remainder, a very considerable portion is represented by the spark loss. The portion remaining unaccounted for is, within limits of operation common to practice, small.

CHAPTER VIII.

THE EFFECT OF THICK FIRING ON BOILER PERFORMANCE.

64. The Conception Underlying these Tests.—It has already been shown (Chapter VI.) that the efficiency of the boiler of a locomotive may vary between wide limits, even when there is no corresponding change in the power delivered. For example, in Fig. 85, Chapter VI., the points representing the results of experiments are in many cases at some distance from the mean curve upon which they should fall. In reviewing the data it has appeared that if the fire had been maintained with an equal degree of efficiency in all of the thirty-five tests reported, every point in Fig. 85, Chapter VI., would have fallen on the curve given, or upon some other curve closely approaching that which is shown. For this reason it has been assumed that the differences in boiler performance noted are effects resulting from changes in the condition of the fire, the extent of which in any individual case is not easily detected by the fireman.

The fireman of a locomotive is guided in his work by the indications of the pressure-gauge rather than by the general condition of the fire. So long as the indication of the gauge is satisfactory but little attention is given the fire, so that whenever the conditions of running are constant and favor easy steaming, the fire passes through a cycle of changes somewhat as follows:

1. Fresh coal is spread on the surface of the fire.
2. The fresh fuel burns rapidly, the temperature of the furnace increases, the pressure responds to the increased activity at the grate, and the gauge goes up.
3. The fire in due time reaches a condition of maximum efficiency, and enters upon a process of decline. It allows the passage of much more air than is needed for complete combustion, but since the engine is running under conditions easily sustained, a fire of even low efficiency is

sufficient to maintain the pressure, and the fireman who watches the gauge sees no occasion for opening the fire-box door.

4. From being thin the fire becomes open and in spots even dead. The influx of air through this open fire increases until its volume is so far in excess of that required for combustion that the efficiency of the furnace falls to a point where it cannot supply the steam required by the cylinders.
5. The hand of the gauge moves downward and the fireman adds new coal, which serves as the starting-point for another round of changes similar to those which have just been described.

The extent to which the routine defined above occurs in practice depends upon the condition of running. Probably the greatest loss from excess air occurs when the conditions of running favor easy steaming, though there is ample evidence to prove that this is not always so. In any case, a remedy is to be found in maintaining the proper thickness of fire, or by checking the draft on a fire that is too thin by means of the ash-pan dampers. Practically, however, the difference between a fire that is too thin and one that is just thick enough is a matter not easily determined.

65. **The Tests and their Results.**—The lack of harmony in the results of the thirty-five tests already presented suggested the considerations mentioned above, and the fact that they well represent conditions of practice emphasized their importance. With a view to demonstrating the losses incident to normal firing, it was determined to run a series of tests at different powers, under conditions which would allow the continuous maintenance of a thick fire, the care of which should be at all times based upon furnace action, rather than upon steam pressure. It was thought that the results of such a series might serve in the location of a curve similar to that of Fig. 91, which represents the average evaporative efficiency as previously defined, and that by avoiding thin fires the points sought would be found above the average curve. In accordance with this conception, tests were outlined upon the following principle:

1. The force of the exhaust producing the draft was to remain constant throughout each test, this condition to be secured by running the engine under a constant speed, load, cut-off, and throttle-opening. The draft itself, as measured

by the vacuum in the smoke-box might vary somewhat,* but the conditions just specified would make the force of the exhaust-jet practically constant.

2. With the exhaust action constant, it was proposed to maintain a heavy fire, and to make all the steam that could possibly be generated. If more was generated than was needed by the cylinders, the excess was to be blown out at the safety-valve.
3. The engines of the locomotive were to have no part in the results, except that of providing a constant draft action; hence the purpose of the test would not be interfered with by wasting steam from the safety-valve.
4. The duration of each test was to be such as would result in the evaporation of not less than 30,000 pounds of water.
5. The draft condition for each test was to be such as could easily be maintained, and so chosen for the different tests that the rates of combustion resulting would cover fairly well the range of conditions common to practice.

It will be seen from the foregoing outline that the fundamental idea was to have the fire always in a condition of maximum efficiency; to have coal applied whenever the condition of the fire made it desirable, instead of waiting until the steam-gauge should prompt the fireman, and to have the fireman watch the fire-box rather than the steam-gauge.

The labor of carrying out the work prescribed by such an outline was undertaken by Mr. O. Harlan, while a graduate student in the laboratory, and the brief summary of results which is here presented has been abstracted from an elaborate thesis presented by him.

The firing was by Charles Reyer, who alone had performed this part in connection with the testing-plant for several years previous to the tests. His instructions were to keep a heavy fire and to burn all the coal that the draft would handle, the expectation being that a full return for all fuel burned would be found in the water evaporated.

The general results are shown in Table XXXIX.

66. Interpretation of the Results.—Plotting the equivalent evaporation with the curve defining the normal performance of the boiler, as determined from the thirty-five tests already discussed, it will be seen

* Slight changes in the condition of the fire itself affect the draft action. (Chapter XI.)

TABLE XXXIX.

Number of test	1	2	3
Month and day in 1897	Feb. 27	April 5	March 6
Duration of test, minutes	360	180	170
Barometric pressure	14.8	14.3	14.6
Fuel.			
Dry coal fired, pounds	5673	6055	6507
Coal per hour	945	2018	2297
Coal per foot of grate per hour	54	115	131
Water and Steam.			
Temperature of feed	52.4	55.3	54.8
Pressure in boiler	128.8	128.4	127.1
Moisture in steam, per cent	1.5	1.4	1.7
Water delivered to the boiler	36228	32734	34500
Water evaporated per pound of coal	6.39	5.41	5.30
Equivalent evaporation per pound of coal	7.42	6.17	6.13
Equivalent evaporation per hour	7011	12451	14080

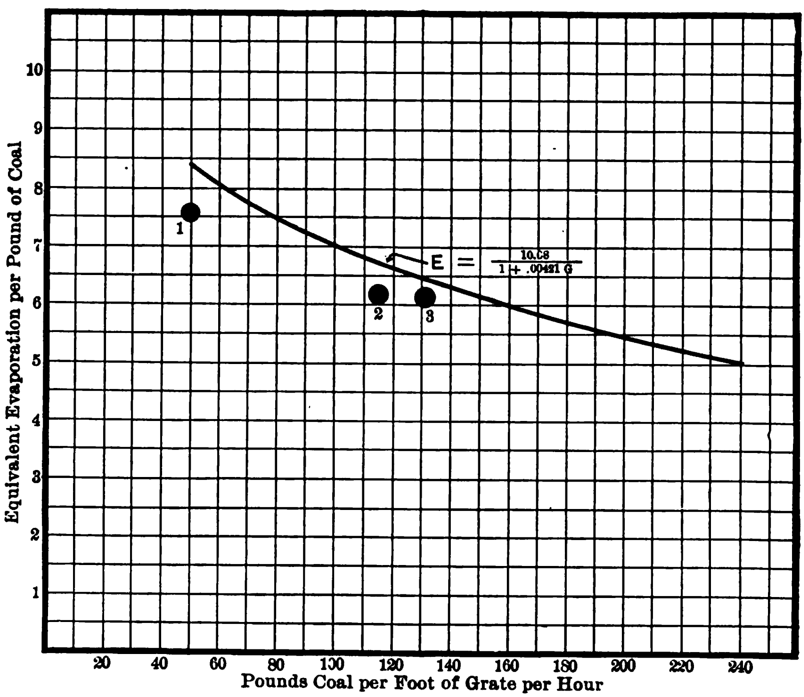

FIG. 91.—Evaporative Efficiency obtained as a Result of Thick Firing as compared with a Curve of Normal Efficiency.

(Fig. 91) that the points all fall low; the loss of evaporative efficiency measured in the per cent of normal efficiency being

for Test No. 1

$$\frac{823-742}{823}100=9.8\%,$$

for test No. 2

$$\frac{681-617}{681}100=9.4\%,$$

for test No. 3

$$\frac{652-613}{652}100=6.0\%.$$

The results are the reverse of those which were expected. They do not, however, prove the non-existence of the defective conditions which it was sought to overcome, but rather that an effort to meet them if pursued too vigorously may lead to worse conditions. A fire that is too thin is bad, but the results of these special tests prove that one which is too thick may be worse.

67. The Influence of the Fireman.—The tests and the results obtained therefrom emphasize the importance of the fireman as a factor in the economical operation of the boiler. As the work of the tests progressed the fireman was impressed with the belief that he was not getting the best possible results, although in carrying out the directions which had been given him he was attentive and painstaking. The results corroborate this opinion, and it is probably true that an experienced fireman can judge with accuracy when his fire is in a condition to make the boiler do its best, but the variations in performance under conditions which are identical, as recorded in the previous chapter, are proof that even though the man be one of exceptional ability and skill no very nice discrimination can be made.

The conclusions to be reached from these tests may be summarized as follows:

1. A fire may be readily maintained so thick as to greatly impair the efficiency of the boiler, probably because of an insufficient supply of air.
2. Between the limits of a very thin and a very thick fire, there probably is, for every condition of draft, a corresponding thickness of fire which will give maximum efficiency.
3. While the tests of the preceding chapter show striking variations in boiler performance under conditions which are

identical, those now under discussion emphasize the importance of the fireman's judgment and skill as factors affecting the efficiency of a locomotive boiler.

4. The results emphasize the difficulty to be met in any attempt to duplicate results from tests of a boiler of a locomotive when fired with coal.
5. The importance of a thorough study of the smoke-box gases in any precise analysis of boiler action is also made evident by the results, a fact which at the time the tests were made had not been fully appreciated.

CHAPTER IX.

SPARK LOSSES.*

68. Sparks.—The passage over the heating surface of a boiler of particles of fuel, more or less consumed, in the form of sparks or cinders, is by no means an inconsiderable source of loss. Experiments for the purpose of determining the extent of this loss were first undertaken in connection with the tests for efficiency at high rates of combustion recorded in Chapter VII., and were afterward continued as a feature of the regular efficiency tests, the results from certain of which are herewith presented.

Solid particles from the fire either lodge in the front end or pass out from the top of the stack. Those which lodge in the front end are sometimes called front-end cinders, those which pass from the stack being sparks; but in this discussion the term sparks will be understood to include all solid matter passing the tubes. It will appear hereafter that the composition of the cinders and sparks may vary from slightly burnt coal to ash. In determining the extent of the fuel loss, resulting from the flight of solid particles from the fire, it is not difficult to ascertain the weight of cinders which lodge in the front end, for they may be readily collected at the end of a test. The determination of the weight of sparks passing out of the stack is a matter of more difficulty. In the experiments in question, this was accomplished by intercepting portions of the stream issuing from the stack, and by collecting the sparks entrained therein. From samples thus obtained, the weight of sparks in the entire stream was estimated. For this work certain special apparatus was designed and constructed which has since been known as a spark-trap.

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* See also "Locomotive Sparks," published by John Wiley & Sons, New York City, for a discussion of the distribution of sparks on either side of the track, and the chance that fires may start therefrom.

69. The Spark-trap (Fig. 92) consisted of an inverted U tube of galvanized iron, securely fastened to a movable frame, by means of which the tip, which constituted one extremity of the tube, could be projected across the top of the locomotive smoke-stack. The outer end of the tube could thus be made to intercept a portion of the stream issuing from the stack, and the continuous action of this stream was sufficient to drive the intercepted portion through the tube and out at the other end. The gases passing the tube bore the sparks on their current, and they were collected in a suitable galvanized iron receptacle set to entrap them. The connection between the tube and

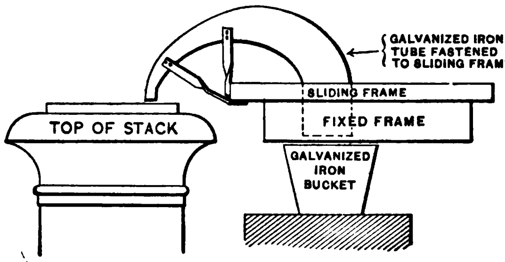


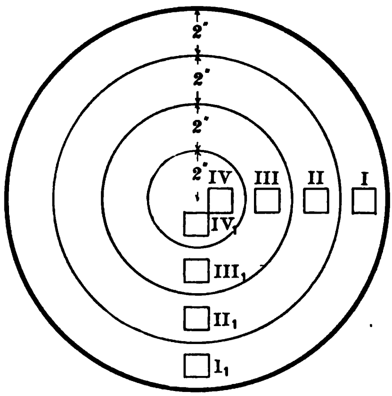
FIG. 92.—Spark-trap.

the receptacle was screened by brass milk-strainer netting. Reference-marks upon the sliding and the fixed frames permitted the tube to be placed in definite locations relative to the center of the stack. This device, when in service, caught everything excepting the lightest soot, which was allowed to escape through the screen unaccounted for.

Assuming the cross-section of the stream issuing from the stack to be cut up, by a series of concentric circles, into one circular and several annular areas, as shown in Fig. 93, the small end of the U tube was placed in the position marked I, and held there for thirty minutes, the sparks collected during this interval being credited to this position. The tube was then moved to the position II, where it remained for another period of thirty minutes. In like manner, it was made to occupy successively the positions III and IV, and also the positions I_1, II_1, III_1, and IV_1, the weight of sparks caught during each interval being credited to the corresponding position occupied by the small end of the tube.

This end of the tube had an area of 2.6 square inches, and it was assumed that the average weight of sparks passing through the tube, while in the positions I and I_1, would be the same as that passing any area of equal extent in the annular space in which these positions are

located. For example, the outer annular area, comprising positions I and I_1, contained 88 square inches. If in half an hour 0.5 of a pound of sparks were caught by the tube in position I, and in another half an hour 0.8 of a pound were collected from the position I_1, the sum of these two weights divided by the area of the sampling-tube (2.6), or .5 of a pound, would be the average weight per square inch per hour collected from the two positions, and the total weight for the annular area would be .5 times 88 or 44 pounds per hour. A similar experiment and calculation gave the weight per hour delivered by each of the other annular areas II and III, and by the circular area IV. The



FIB. 93.—Plan of Top of Stack.

sum of these separate determinations was assumed to be the total weight of sparks per hour delivered from the stack.

70. Conditions of Tests.—The series of seven tests for which spark losses were determined embrace a wide range of conditions, the speed varying from fifteen to fifty-five miles per hour, the cut-off from twenty-five per cent to eighty per cent of stroke, the draft from two inches to five inches of water pressure, and the rate of combustion from forty-five to one hundred and twenty pounds of coal per square foot of grate surface per hour. It will thus be seen that the results should fairly represent common practice. All parts of the boiler and engine were in normal condition, and each test was conducted at constant speed and load for a sufficient time to permit accuracy in securing samples of sparks. The boiler, grate, and front-end arrangement employed during the tests are shown by the drawings presented with Chapter III. The exhaust-nozzle was double, three inches in diame-

ter, and the stack was sixteen inches in diameter. As in all other tests herein recorded, the coal used was Brazil block.

71. Observed Weight of Sparks.—Table XL. gives a summary of the observed data with the tests arranged in order of rates of combustion (Column 5), Test No. 1 having the lowest rate. The laboratory symbol (Column 2) is given to enable the tests to be identified in the general presentation of data (Chapter IV.), in case it is desired to compare boiler performance or other facts. The weight of sparks passing from the stack per hour and the weight caught in the front end per hour are given in Columns 6 and 7 respectively, while the sum of these two appears in Column 8. Column 9 gives the ratio of total weight of sparks to total weight of coal.

TABLE XL.

OBSERVED VALUES.

Number.	Laboratory Symbol.	Draft.	Coal Fired per Hour. Lbs.	Coal Fired per Square Foot of Grate per Hour. Lbs.	Sparks Passing out of Stack per Hour. Lbs.	Sparks Caught in Front End per Hour. Lbs.	Total Sparks per Hour. Lbs.	Ratio of Total Weight of Sparks to Weight of Coal Fired.
1	2	3	4	5	6	7	8	9
1	15-1-H	1.93	791.5	45.23	21.9	12.3	34.2	.043
2	35-1b-G	2.98	1464.8	83.70	41.2	63.2	104.4	.071
3	35-1-G	3.02	1513.6	86.50	46.1	49.9	96.0	.063
4	35-1-H	3.00	1557.1	88.98	45.0	63.4	108.4	.070
5	55-1-G	3.57	1884.4	107.68	127.5	157.2	284.7	.151
6	35-3-G	4.88	2017.7	115.29	110.0	67.3	177.3	.088
7	15-9-H	4.99	2121.0	121.20	215.5	78.0	293.5	.138

Values given in Columns 8 and 9 agree closely with similar values obtained from the special tests recorded in the preceding chapter. Thus, both series of tests show large increase in the spark loss, with increase in rate of combustion, losses from this source for the tests under consideration being from four to fourteen per cent of the total weight of coal fired. Unlike the results which are presented in a preceding chapter, they were obtained under conditions common to everyday service.

A graphical representation of the total spark losses, of the proportion caught in the front end, and of the amount going out of the stack for the several tests, is presented as Fig. 94. The shaded portions represent the sparks caught in the smoke-box, the light portions those passing out of the stack. Referring to this figure it will be seen that in general

the total weight of sparks increases with the rate of combustion and draft, and that the proportion of sparks passing out of the stack, as compared with those remaining in the smoke-box, also increases. Test No. 1 is an exception to the latter statement, but inasmuch as the total loss in this case is small, the record may not be entirely trustworthy. In Test No. 2 the stack losses are less than half, while in Test No. 7 they constitute more than two-thirds of the total loss. The results of Tests Nos. 2, 3, and 4 are of especial interest, since they were run under very similar conditions, and should therefore agree closely. Test No. 5 shows a smoke-box loss relatively higher than that of either Tests Nos. 6 or 7. This result, while unexpected, may be accounted for

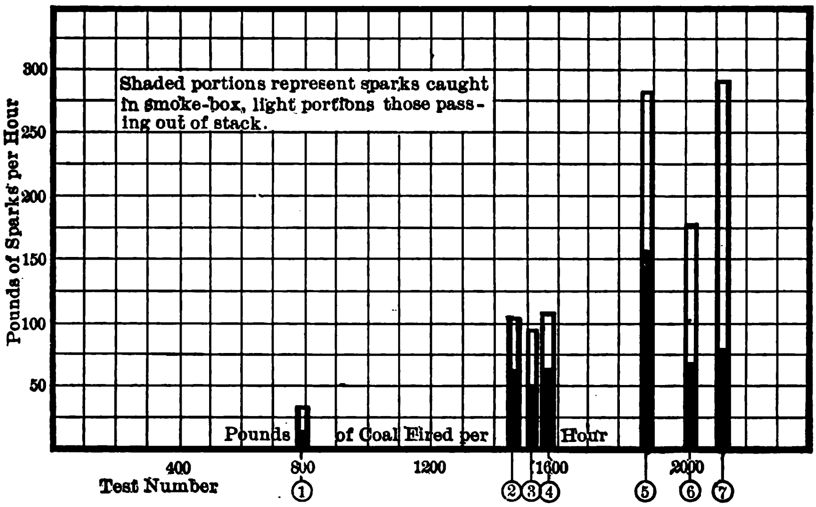


FIG. 94.—Spark Losses.

by the fact that this was a high-speed test. Test No. 6 may be more nearly comparable with Test No. 3 than with Test No. 5, owing to the high speed of the latter test. The smaller values given for Test No. 6, as compared with those derived from Test No. 5, if not the result of inaccuracies, must be due to the fact that the cut-off for this test was considerably increased, the conclusion being that while the draft produced was stronger and more coal was burned, the effect of the exhaust-jet was less violent than that of the more rapid blast at high speed. The data, however, is too meager to be convincing upon this point, and the conclusion suggested is not in harmony with a conclusion reached as a result of a study of the action of the exhaust-jet (Chapter IX.).

The conclusion referred to is to the effect that draft action is independent of speed. Test No. 7 was run under a long cut-off with the throttle partly closed and at low speed, the condition being such as to give a very strong exhaust action. The draft was greater than for any of the preceding tests, and more coal was burned per unit of time. The weight of sparks caught in the smoke-box is not greatly in excess of the weight caught in any of the preceding tests and is only about half that for Test No. 5, but the weight passing out of the stack is much greater than for any other tests; the strong blast evidently tending to clear the front end of a portion of the accumulation which otherwise would have lodged there.

As the total spark loss increased the relative proportion passing from the stack increased, a result which was probably due in part to the strong scouring action incidental to heavy draft and to the limited capacity of the front end. It is evident, also, that the length of the test will have a decided influence on the distribution of the sparks. Thus, in the case of a long test the front end may become completely filled, after which all solid particles must pass up the stack.

72. **The Heating Value of the Sparks** is disclosed by Items 4 and 5 in Table XLI.

TABLE XLI.

EQUIVALENT COAL.

Number.	Coal Fired per Hour. Lbs.	Total Sparks per Hour. Lbs.	Pounds of Coal Equivalent in Heating Value to One Pound of Sparks.	Pounds of Coal Equivalent in Heating Value to Sparks per Hour.	Per Cent of Fuel Accounted for as Sparks.
1	2	3	4	5	6
1	791.5	34.2	.665	22.7	2.9
2	1464.8	104.4	.801	83.6	5.7
3	1513.6	96.0	.807	77.5	5.1
4	1557.1	108.4	.812	88.0	5.7
5	1884.4	284.7	.841	239.4	12.7
6	2017.7	177.3	.851	150.9	7.5
7	2121.0	293.5	.856	251.2	11.9

The fuel values in equivalent pounds of coal per pound of sparks recorded in this table were obtained in the following manner: The results of numerous analyses of sparks were plotted against the rates of combustion at which they were produced, and a smooth curve was drawn as nearly as possible through the points. This curve is reproduced as Fig. 95. As the experiments show that equal rates

of combustion will produce sparks of equal heating value, the values of Item 4 were read from the curve. Multiplying Item 4 by Item 3, the weight of coal equivalent to the spark loss per hour, Item 5 was obtained.

The values of the table confirm a conclusion previously stated, namely, that as the power and rate of combustion increase, both the total quantity of the sparks and their heating value per pound increase. Thus in Test No. 1 the weight of sparks was about 4 per

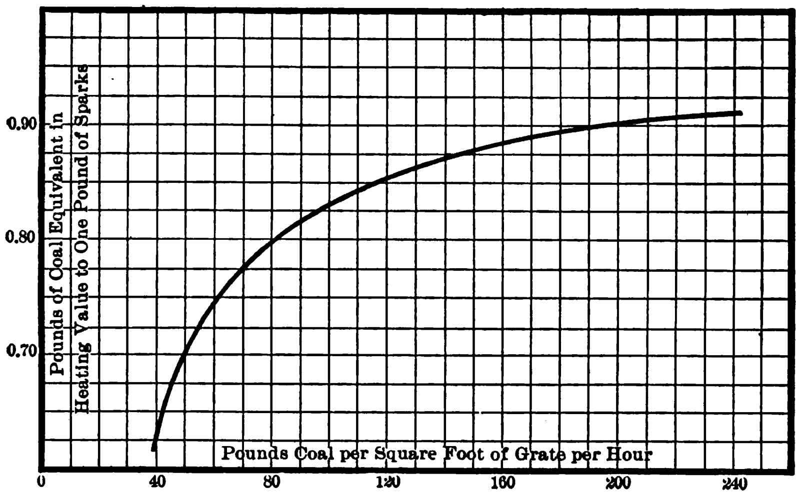


FIG. 95.—Heating Value of Sparks.

cent of the weight of the coal fired, and a pound of the sparks was equivalent only to four-sixths of a pound of coal, whereas, in Test No. 7, when the engine was working hard, about 14 per cent by weight of the coal escaped as sparks, and each pound of sparks was equivalent to nearly five-sixths of a pound of coal. Under the conditions of this test, therefore, more than 10 per cent of the fuel which enters the furnace may completely pass the heating surface unconsumed.

73. Volume of Sparks as Dependent upon Quality of Fuel.—In 1898, in connection with locomotive Schenectady No. 2, a series of tests designed to determine the relative value of five different samples of coal were undertaken by the Engineering Laboratory for the C. C. C. & St. L. Railway Company, in connection with which the extent of spark losses was determined. The several samples of coal were designated as *A*, *B*, *C*, *D*, and *E*, and the evaporative efficiency

obtained from them in locomotive service is that set forth in Table XLII., the values of which are arranged in the order of merit. From the evaporation obtained the value of the several samples may be estimated. Sample *E* was a West Virginia coal of very high quality, while sample *C* was a Western coal, probably from an Indiana or Illinois mine.

TABLE XLII.

EQUIVALENT EVAPORATION.

Designation of Sample.	Pounds of Water Evaporated per Pound of Coal.		Relative Value of Sample, calling the Value of Sample E 100.	
	When the Rate of Evaporation is 5.	When the Rate of Evaporation is 10.	For Use under Light Power.	For Use under Heavy Power.
I.	II.	III.	IV.	V.
E	10.7	8.7	100	100
A	10.0	8.1	93	93
B	9.4	7.7	87	89
D	8.3	7.0	79	82
C	8.5	7.1	77	81

The spark loss is assumed to be the weight caught in the front end plus that which passes out of the stack. Values for spark losses thus obtained for the several samples of coal tested, expressed as a percentage of the weight of coal fired, are presented in Columns II. and III. of the table below.

TABLE XLIII.

SPARK LOSSES.

Designation of Sample.	Percentage of Weight of Coal Fired, Accounted for as Sparks Entrapped in Front End and Passing out of the Stack.		Relative Weight of Sparks Produced by each of the Several Samples in Generating the Same Weight of Steam, assuming the Weight of Sparks resulting from Sample E to equal 100.	
	When the Rate of Evaporation is 5.	When the Rate of Evaporation is 10.	When the Rate of Evaporation is 5.	When the Rate of Evaporation is 10.
I.	II.	III.	IV.	V.
E	6.6	21.2	100	100
A	6.2	15.6	101	79
B	4.8	16.4	84	87
D	5.7	17.4	109	100
C	3.3	13.8	65	80

From the preceding table it appears that under conditions of running common in every-day service, from 14 to 21 per cent of the coal fired disappears in the form of sparks (Column III). It is true, however, that the fuel loss is not as great as this, since the sparks represent fuel which has been partially consumed. The sparks resulting from the samples under consideration have not been analyzed, but the investigations already described indicate that they have from 60 to 85 per cent of the fuel value of a similar weight of coal. A fair estimate of the fuel losses in these tests, resulting from the passage of sparks through the tube, may be taken as 75 per cent of the values given in Columns II. and III.

From a review of the tables it appears that those samples giving the highest evaporation also give the largest spark losses. Two conditions probably account for this fact. First, the purer coals are lighter, and hence respond to the draft action more easily than those intermixed with non-combustible matter; secondly, in general, the better the coal the lighter the ash, a large percentage of which, instead of falling through the grate passes out with the sparks, and adds its mass to their weight.

It may be urged as an objection to those coals giving a high efficiency, that their use is attended by a large spark loss. The argument, while good, is not true to the extent indicated by the values in Columns II. and III. For example, Column III. shows the percentage of the coal fired, accounted for as sparks, but a pound of sample *C*, producing 0.138 pound of sparks, did not make as much steam as a pound of sample *E*, producing 0.212 pound of sparks. The relative spark-producing qualities of the several samples, based *upon the weight of steam generated*, are given in Columns IV. and V. These values, therefore, serve as a logical basis from which to determine the relative spark-producing qualities of the several samples.

74. Refuse Caught in the Ash-pan.—Closely allied with the matter of spark production is that of refuse in the ash-pan, since, as already noted, when the ash is light much of it passes up the stack. The facts with reference to ash, as applying to the five samples of coal dealt with in the preceding paragraph, are given in Table XLIV.

The table shows that when the engine is running light, the five samples give nearly the same amount of refuse in the ash-pan, whereas, when the power is increased to make the rate of evaporation 10, sample *C* gives nearly three times as much deposit in the ash-pan as sample *E*. In general it may be said that the better the fuel, the less deposit

there will be in the ash-pan. Comparing the values of Table XLIV. with those of Table XLIII. it appears that as the character of the coal changes with reference to refuse in the ash-pan, an inverse change results with reference to spark losses. Such a result is logical, and is quite in accord with the explanation given in the discussion of spark losses.

TABLE XLIV.

REFUSE IN ASH-PAN.

Designation of Sample.	Percentage of Weight of Coal Fired, Accounted for as Refuse in Ash-pan.		Relative Weight of Ash resulting from Different Samples when the Same Weight of Steam is Generated, calling the Weight resulting from Sample E 100.	
	When the Rate of Evaporation is 5.	When the Rate of Evaporation is 10.	When the Rate of Evaporation is 5.	When the Rate of Evaporation is 10.
I.	II.	III.	IV.	V.
E	11.0	4.9	100	100
A	10.7	9.2	105	203
B	13.9	11.6	146	267
D	14.8	10.3	170	257
C	14.4	11.0	171	276

75. Distribution of Sparks Throughout the Stack.—To intercept the sparks at some point between the front tube-sheet and the top of the stack is one of the problems which confront designers of draft appliances. The development of a successful device for the purpose is likely to be advanced by a knowledge of the course taken by the sparks in leaving the boiler. It is therefore thought proper to present here whatever information may have been procured in the course of this investigation with regard to spark distribution in the stack. For this purpose the data of Test No. 4, which are considered to be representative, are presented graphically in Figs. 96, 97, and 98. In Fig. 96 the numbers express the weight of sparks passing out of the stack per square inch per hour at the places designated, while in Fig. 97 they indicate he total pounds of sparks per hour passing each of the several areas into which the stack was arbitrarily divided.

Comparing these results it appears that the sparks follow the outside of the exhaust steam rather than the center. The weight of sparks per unit area increases steadily from the center to the circumference, with the result that over 50 per cent of the whole weight is credited to a ring two inches broad, measured from the outside circum-

ference, which ring contains only about 40 per cent of the total area. The velocity of the exhaust is necessarily less on the outside of the stream, and, apparently, the sparks most readily follow the portion of stream issuing from the stack with the least velocity. In this connection it should be noted that these observations were all made on a cross-section of the stream as it issued from the stack, and therefore do not necessarily represent conditions actually existing in the stack.

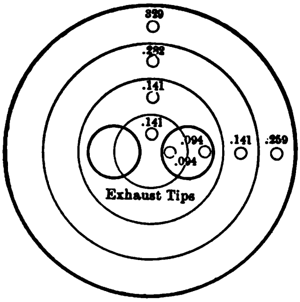


FIG. 96.—Pounds of Sparks passing out of Stack per Square Inch per Hour.

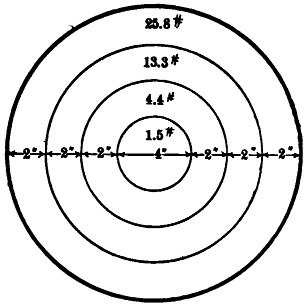


FIG. 97.—Pounds of Sparks passing out of Stack per Hour in the Areas Indicated.

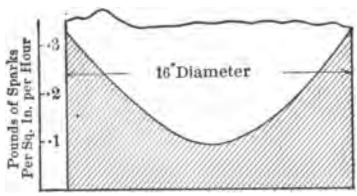


FIG. 98.—Cross-section of Stack showing Density of Spark Discharge.

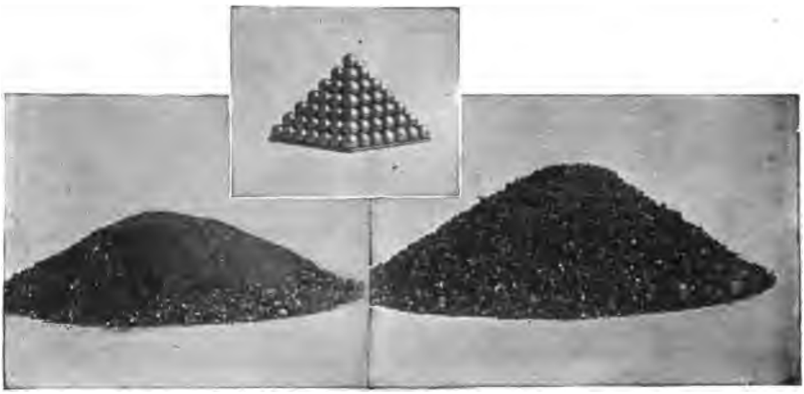
76. The Size of Sparks varies with the extent of the spark losses. Thus, under low rates of combustion, when the total spark loss is small, it consists of a very fine, almost sooty, deposit (*A*, Fig. 99). but when the total loss becomes large, the sparks themselves are large (*B*, Fig. 99). Fig. 99 is a photograph of two lots of sparks, and of a pile of buckshot, with which the sparks may be compared. The sample *A* was obtained in Test No. 1, when the total loss was only 22 pounds per hour, while the sample *B* represents Test No. 7, for which the loss was more than ten times as great.

77. Conclusion.—The results of the investigation described in the preceding paragraphs seem to justify certain conclusions which, while

susceptible of rather general application, should nevertheless be accepted only for the fuels and boilers involved. Both locomotives involved, Schenectady No. 1 and No. 2, have narrow fire-boxes and a grate area of about 17.5 square feet. The conclusions are as follows :

1. The weight of sparks, which passes the heating surface of a boiler as cinders or sparks, increases steadily with the rate of combustion, and may reach a value of from 10 to 15 per cent of the coal fired.

2. Sparks are composed of coals more or less completely burned, or they may be wholly non-combustible, in which case they are entirely of ash.



A *B*

FIG. 99.—Sample Sparks.

3. The fuel value per pound of sparks increases as the total amount increases.

4. The weight of sparks produced and the weight of refuse caught in the ash-pan are inversely related. When the spark production is high, much of the refuse which would otherwise drop into the ash-pan passes out from the top of the stack.

5. The distribution of sparks throughout the cross-section of the stream issuing from the stack is such that the greatest weight of sparks follows the slowest currents on the edges of the stream, more than one-half the total weight passing through the annular area which comprises the two inches nearest the stack.

6. The size of the individual sparks increases with the total amount produced up to the limit allowed by the openings in the netting.

CHAPTER X.

RADIATION LOSSES.

78. **The Amount of Heat Radiated** from a locomotive boiler is necessarily large, but few attempts have been made to measure it. It is chiefly for this reason that the experiments herein described are of interest.

79. **Loss of Heat from a Locomotive Standing in a Building.**—By means of a Bristol recording pressure-gauge attached to the boiler of locomotive Schenectady No. 1 many charts were obtained, showing the rate at which the boiler pressure declined after the locomotive had been shut down for the day. These charts indicate that ordinarily the pressure would fall from 120 to 0 pounds in from 12 to 15 hours, the exact time, of course, depending upon many different conditions.

On January 25, 1895, at the close of a test, the water level was brought to the 5″ mark on the glass, and the pressure noted. The chart on the Bristol gauge gave the subsequent record of time and pressure. From the known dimensions of the boiler and these data, the following was readily derived:

Water in boiler, 140.6 cu. ft.	= 8787.5 lbs.
Steam in boiler at 110 lbs. absolute pressure, 53.0 cu. ft.	= 13.2 lbs.
Total in boiler.	8800.7 lbs.
Total weight × heat of liquid.	= 2,685,974 B. T. U.
Weight of steam × heat of vaporization.	= 11,565 B. T. U.

The heat of the steam is but .43 of one per cent of the heat of the liquid, and is negligible. In determining heat losses it will be sufficient to assume that the thermal units dissipated during any given interval are the product of 8800 and the fall in temperature in that interval. Results thus derived are given in Table XLV.

TABLE XLV.

HEAT LOST FROM LOCOMOTIVE BOILER IN BUILDING.

AVERAGE TEMPERATURE OF BUILDING, 71.5° F.

Time.	Absolute Pressure, Lbs.	Temperature, Degrees F.	Fall in Temperature, Degrees F.	Fall in Temperature per Hour, Degrees F.	B.T.U. Lost per Hour.
5.07	110	334.56			
5.20½	100	327.58	6.98	31.02	273,007
5.37½	90	320.04	7.45	26.64	234,459
5.59	80	311.80	8.24	22.98	202,247
6.25½	70	302.71	9.09	20.58	181,125
6.59	60	292.51	10.20	18.24	160,530
7.40	50	280.85	11.66	17.04	149,969
8.36	40	267.13	13.72	14.70	129,374
9.52	30	250.27	16.86	13.14	115,645
11.45	20	227.95	22.32	11.88	104,556

The losses shown have by implication been referred to as radiation losses, but they really include heat losses arising from all sources. A consideration of the methods employed will make it evident that they include losses resulting from leakage either of steam or water. The boiler itself is known to have been tight, and the throttle and other valves are thought to have been so, but experience proves that in so large and complicated a structure as a locomotive boiler and its fittings it is impossible to absolutely avoid leakage. The locomotive was in an exceptionally good condition, and the values given, are to be accepted as representing such losses as occur from causes stated under most favorable conditions.

The coal equivalent of the heat lost may be found by assuming the boiler to absorb 8000 B.T.U. per pound of coal. This assumption is justified by the facts presented in Chapter VI. By its use the heat loss per hour when the gauge shows 95 pounds pressure (absolute 110) is equivalent to

$$\frac{273007}{8000} = 34.1 \text{ pounds of coal.}$$

As the pressure and temperature fall, the heat loss diminishes.

80. Radiation Losses upon the Road.—Tests to determine the heat losses from the boiler of a locomotive on the road were undertaken in coöperation with the Chicago & Northwestern Railroad Company and several manufacturers of boiler-covering in 1898. The immediate purpose of the test was to disclose the relative value of different

materials employed in lagging boilers, but the plan was sufficiently broad to make the results of value in a discussion of the more general subject of heat losses on the road.*

81. Plan of the Tests.—In carrying out the tests two locomotives were employed; one to be hereafter referred to as the "experimental locomotive" was subject to the varying conditions of the tests; the other was at all times under normal conditions serving to give motion to the experimental locomotive, and as a source of supply from which steam could be drawn for use in maintaining the experimental boiler at the desired temperature. The experimental locomotive was coupled ahead of the normal engine, and, consequently, was first, when running, to enter the undisturbed air. The action of the air-currents upon

FIG. 100.—Head of Experimental Train.

it, therefore, was in every way similar to those affecting an engine doing ordinary work at the head of a train.

The boiler of the experimental locomotive was kept under a steam pressure of 150 pounds by a supply of steam drawn from the boiler of the normal engine in the rear. There was no fire in the experimental boiler. It was at all times practically void of water. Precautions were taken which justified the assumption that all water of condensation collecting in the experimental boiler was the result of radiation of heat from its exterior surface. This water of condensation was collected and weighed, thus serving as a means from which to calculate

* These tests were outlined and conducted by the author under the direction of the Chicago & Northwestern Railroad Company, as represented by Mr. Robert Quayle, Superintendent of Motive Power, in coöperation with the manufacturers of various materials employed as boiler-coverings. See also "Proceedings of Western Railway Club," January, 1899.

the amount of heat radiated. The head of the experimental train is shown by Fig. 100.

82. The Experimental Boiler and its Equipment.—The Chicago & Northwestern locomotive, No. 626, the boiler of which served in the experiments, is of the eight-wheeled type, weighing about 90,000 pounds. An outline drawing, used in ordering covering, is shown by Fig. 101. The principal dimensions of the boiler are as follows:

TABLE XLVI.

DIMENSIONS OF BOILER.

Diameter in inches	52
Heating surface (square feet)	1391
Total area of exterior surface, not including surface of smoke-box	358
Area of surface covered (square feet)	219
Area of steam-heated exposed surface not covered	139
Ratio of surface covered to total surface	.61

It should be noted that the values given in Table XLVI. are based upon projected areas of the plain boiler. No account has been made of the edges of plates at joints, nor of surface due to the projection of rivet-heads, nor of the surface of various attached projections, such as running-board brackets and frame fastenings. While all such projections above the general surface of the boiler are active agents in conducting heat from the interior, the present study does not require them to be taken into account. The extent of area covered for this boiler is entirely normal for the class of locomotives to which No. 626 belongs, which gives added interest to the fact that but 61 per cent of the exposed surface of the boiler was covered.

The whole grate of the experimental boiler was deadened by brickwork, and as a further precaution against the movement of air-currents through the fire-box, tubes, etc., the top of the stack was securely filled with wood. The furnace and front-end doors were also carefully closed and fastened. A steam-separator in the supply-pipe within the cab of the experimental boiler was assumed to deliver to it steam of a uniform quality. These and other precautions justify the assumption that all condensation occurring in the boiler was due to radiation from its exterior surface.

As a safeguard against air-pockets and to further insure a uniform temperature of all portions of the interior of the boiler, steam was allowed to waste from it through a small orifice at the end of a pipe connecting with the front end, and leading outside to the top of the

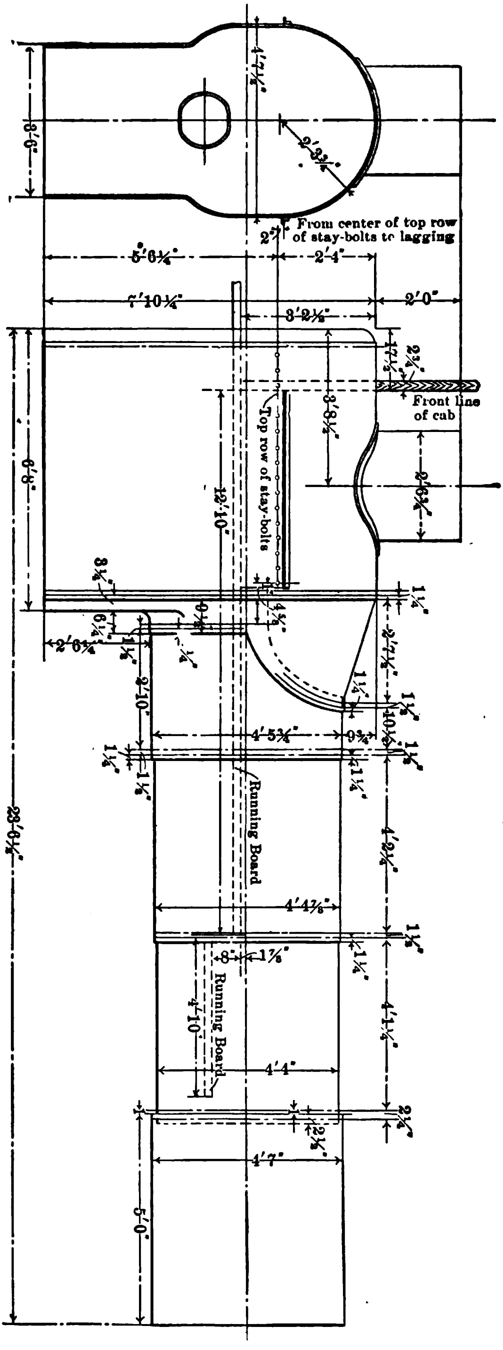


FIG. 101.—Experimental Boiler.

stack, and some leak was allowed also at the whistle-valve. The loss of steam from the experimental boiler in no way affected the value of the measurements made, since they neither increased nor diminished the amount of condensation taking place within the boiler.

A 12-inch water-glass was attached to the water-leg of the boiler close to the mud-ring. A thread around this glass served as a reference-line. The water condensing within the experimental boiler was led through a ¾-inch pipe from the blow-off cock at the bottom of the boiler to a valve at the rear of the cab, thence to the top of the tender-tank, at which point it connected with a coil submerged in the water of the tank. The discharge from this coil was delivered to a weighing-barrel set up within the coal space of the tender. By these means the water of condensation was made a measure of the amount of heat radiated, its level was maintained constant a few inches above the level of the mud-ring, the excess was drained out, cooled to avoid all chance of loss from vaporization, and weighed.

As the scales could not be balanced during a run, and as the weighing-tank was of insufficient capacity to hold all of the water accumulating during a test, a calibrated small-necked can was used between stops to reduce the level of the water in the barrel. Each can, as emptied, was charged against the barrel, a full can counting 45.7 pounds.

83. **Observers.**—Three observers were ordinarily employed during each test. One was assigned the duty of observing the force and direction of the wind, the character of the weather, and to so manipulate the valve in the pipe through which the condensed steam was discharged to the weighing-barrel, that the water level within the experimental boiler would at all times remain near the reference-line. Another weighed the condensed steam, and a third recorded five-minute readings of the steam pressure within the experimental boiler, and attended to the discharge drain of the steam separator, in order that the water level within this apparatus might be kept within fixed limits. He also rang the locomotive bell for crossings. The rear engine carried its usual crew. The train conductor also rode on the rear engine.

84. **The Track** used for the running tests is a single line extending from Clinton to Anamosa, Iowa, a distance of seventy-two miles. This stretch of road was chosen because of the light traffic upon it, and the assurance against interruption which this condition gave. It leads over rolling country, and throughout its length is rather

sinuous. For the first twenty-five or thirty miles the general direction is northwesterly, and for the remainder of the distance nearly west. For four miles out of Clinton it extends through the yards of that city and the adjoining city of Lyons, and for several miles it follows along the Mississippi River, from which it finally leads out upon a more open country. The wind and temperature conditions were generally different for that portion of the road along the river than for portions extending across the more open country. Crossing stops were necessary just beyond Lyons and at Delmar, thirty-three miles from Clinton. There are fifteen stations intermediate between terminals, but, with one exception, the plan of the tests did not involve them.

85. Movement During the Tests.—The work in connection with each covering occupied ordinarily a single day. A test under conditions of rest, hereafter referred to as a "standing test," was first made, followed by a test on the road, hereafter to be referred to as a "running test."

Each running test involved a trip from Clinton to Anamosa and return, with an intermediate stop each way at the station of Maquoketa, thirty-eight miles from Clinton and thirty-four miles from Anamosa. By means of these stops it was possible to divide each running test into four parts of nearly equal length, which not only gave opportunity for ascertaining something of the character of the results, which were being obtained, but served as a safeguard against the loss of a whole test in case of an accident on the road. For convenience these parts of the running tests are hereafter referred to as "quarters," but they are not of equal value. The quarters may be defined as follows:

1st quarter, Clinton to Maquoketa, 38 miles, approximate running time, 83 minutes.

2d quarter, Maquoketa to Anamosa, 34 miles, approximate running time, 68 minutes.

3d quarter, Anamosa to Maquoketa, 34 miles, approximate running time, 71 minutes.

4th quarter, Maquoketa to Clinton, 38 miles, approximate running time, 84 minutes.

In anticipation of a test the locomotives were coupled, the pipe connections made, and steam turned on the experimental boiler at as early an hour as practicable. In most cases this was between 6 and 7 o'clock in the morning. After the normal pressure had been secured in the experimental boiler, the cooling coil within the tender was put

under pressure and examined for leaks. The engines were then moved to the yard stand-pipe and the tender-tanks of both engines filled. As soon as practicable after this, and to insure the same level of the experimental boiler for all tests, the engines were moved to a point lower down in the yard where the rear driver of the experimental engine rested over a certain marked tie. The water within the experimental boiler, resulting from condensation, was then brought to the reference-line, time was taken, and the scales of the weighing-barrel balanced. At fifteen-minute intervals thereafter this process of bringing the water to line and balancing the scales was repeated, usually for a period of from one to two hours, the locomotive remaining in its place upon the marked tie. When the rate of condensation became uniform the standing test was assumed to have commenced.

In due time, usually at about 9:35 in the morning, the water was brought to line for the last observation of the standing test, the scales balanced, and as soon as practicable thereafter the engines were started for the running test. The time of this balancing of the weighing-tanks marked the end of the standing test and the beginning of the running test.

The first few miles were, necessarily, at varying speed, but after passing Lyons and the crossing just beyond, a speed of thirty miles an hour was soon secured, and was thereafter maintained until Maquoketa was reached. In all tests the stop at Maquoketa was made with the rear tender under the spout of the water-tank, water was taken by the pushing engine, and the engines were oiled. While this was being done the water of condensation in the experimental boiler was brought to line and the weighing-barrel balanced, thus ending the first quarter and beginning the second quarter of the running test. After a ten minutes' stop start was again made and the run continued to Anamosa, where the stop was made with the rear driver of the experimental locomotive on a certain marked tie. As soon as practicable thereafter the water was brought to line and the weighing-barrel balanced, thus ending the second quarter of the test.

At Anamosa the engines were uncoupled and turned one at a time, coupled again, and the rear driver of the experimental boiler brought over the same marked tie upon which stop had been made. Steam was then turned on the experimental boiler, and the pressure which, during the process of turning, usually dropped to about eighty pounds, was restored to normal conditions. The start from Anamosa was made at about 1:45, or an hour and a half after the scheduled time

of arriving. This interval gave time for the work of balancing the weighing-barrel, turning the engines, and for dinner; it also gave a sufficient period, after normal pressure had been restored in the experimental boiler, to allow everything to become thoroughly warm before starting.

When all was ready the condensed steam in the boiler was brought to line, the weighing-tank balanced, and the third quarter of the test thus commenced. As soon as practicable after the balancing, the train was started on the run to Maquoketa, where, as before, the stop of ten minutes' duration was at the water-tank. Here, again, the water was brought to line and the weighing-tank balanced, thus ending the third quarter and beginning the fourth quarter of the test, which in turn ended at Clinton at about 4:30 in the afternoon upon the same marked tie from which the engines had been started in the morning.

This process was persisted in with regularity, the intent being to secure similar conditions for each of the several tests.

After the final balance the tank-valves of the experimental locomotive were opened and the tank drained to allow the inspection of the cooling coil, which inspection, as already stated, preceded every test. Steam was shut off the boiler and the experimental locomotive was pushed into the roundhouse, where men were in waiting to strip it of its jacket and covering. Early in the evening work was commenced in applying the covering which was to be tested on the following day, and was continued into the night until finished.

It was found impossible to make the running time of all tests the same. Conditions arose which could not have been anticipated. There were occasional stops due to section gangs and to the presence of other trains. Time which was lost in this way was not made up during the run, the effort being to keep the *speed while running constant*. It is to be noted also that the running tests actually commenced before the train was started, that the ten minutes' stop at Maquoketa was a part of the running time, and that the test did not end the moment the engine stopped, that is, there was a certain amount of dead time on all running tests. The facts in detail with reference to this are fully presented in another place.

86. The Coverings Tested.—Tests were made in conjunction with the bare boiler (A), with an old covering of wood (B) which was on the boiler when the work began, and with five different forms of manufactured coverings designated as C, D, E, F, and G respectively. The list included all of those materials now common as insulating materials,

as, for example, magnesia, asbestos, and cellular asbestos board. The thickness in all cases was designed to be the same for all tests, but the practice of covering the surface of a locomotive boiler, with its inequalities, with material in such thickness as will give a smooth exterior surface, leads to variations in the thickness of the covering. In the boiler tested the dome-casing was so small as to allow only a thin layer (from $\frac{1}{2}$ to $\frac{3}{4}$ in.) on the barrel. On other portions of the boiler it was intended that the thickness should vary from $1\frac{3}{4}$ in. on the first ring to $1\frac{1}{4}$ in. on the slope sheet. An effort was made to have the thickness of all coverings tested the same, and, except in two cases, this result was very nearly attained.

As a comparative figure an average thickness of all coverings was obtained by finding the volume of the material used, and dividing this by the area of the surface covered. This process gives the thickness which the material would have had if it had been distributed uniformly over the surface covered. Values thus obtained are as follows:

Covering	B	1.34 in. $= 1\frac{11}{32}$ in.
"	C	1.49 in. $= 1\frac{16}{32}$ in.
"	D_1	1.45 in. $= 1\frac{15}{32}$ in.
"	D_2	1.57 in. $= 1\frac{18}{32}$ in.
"	E	1.28 in. $= 1\frac{9}{32}$ in.
"	F_1 and F_2	1.56 in. $= 1\frac{18}{32}$ in.
"	G	1.49 in. $= 1\frac{16}{32}$ in.

In all cases the material as above described was covered by the usual Russia iron lagging.

87. **The Tests.**—Both standing and running tests were made with the experimental boiler bare, and also when protected by six different coverings. Tests of two of these were repeated, making altogether nine standing tests and nine running tests to be reported. These are designated as follows: A, B, C, D_1, D_2, E, F_1, F_2, and G. "A" represents the test of the bare boiler. "D_1" and "D_2" are different tests of the same covering, and, similarly, "F_1" and "F_2" are tests of a single covering.

88. **Standing Tests and Results.**—All standing tests, save one, immediately preceded the corresponding running test. Standing Test C followed the running test.

The observed data, and the results which have been derived from them, are presented as Table XLVII. Most of the lines in this table are self-explanatory, but there are a few which demand a word of explanation.

TABLE XLVII.

STANDING TESTS.

1	Designation of test	Bare A. Boiler.	B.	C.	D_1.	D_2.	E.	F_1.	F_2.	G.
2	Date: month and day, 1898	Aug. 11	Aug. 10	Aug. 12	Aug. 14	Aug. 18	Aug. 15	Aug. 16	Aug. 17	Aug. 19
3	Duration of preliminary warming, minutes	50	93	204	127	111	49	53	75	71
4	Duration of test, minutes	62.3	35.5	30.5	58.9	38.0	26.5	60.3	75.8	40.8
5	Average boiler pressure, pounds	152	156.9	156	150.2	150.1	153.2	152.5	151.5	152.9
6	Average atmospheric temperature, deg. F.	74	72	78	72	72	73.5	71	79	70
7	Direction and force of wind, miles per hour. N	6 to 10	0 to 2	3 to 8	5	3 to 5	5	10 to 15	3	3
8	Observed condensation for test, pounds	432.6	97	106.1	176	109.5	96	189.5	246.5	128.5
9	Amount to be added to or subtracted from 8 to convert it to an equivalent amount, which would have been observed had the average steam pressure been 150 lbs. and atmospheric temperature 80° F.	−10.4	−3.7	−1.8	−4.8	−3.0	−2.6	−6.4	−2.5	−5.0
10	Total condensation under standard conditions of pressure and temperature	422.2	93.3	104.3	171.2	106.5	93.4	183.1	244	123.5
11	Condensation per minute	6.78	2.63*	3.42	2.91	2.80	3.52	3.04	3.22	3.03

* The value 2.63 is probably too low; it is estimated that the true value is not less than 3.

No attempt has been made to correct results for different conditions of wind, but there can be no doubt that changes in exposure arising from this cause had a pronounced effect upon the results of the standing test. For this reason the results of the standing test are believed to be far less reliable for purposes of comparison than those derived from the running tests (a description of which is to follow), for in the running tests the effect of slight variations in wind velocity were in the effect swallowed up of the forward movement of the experimental boiler itself.

The results given for Test B (2.63 pounds per minute) are so low as to be fairly open to question. This was the first test made. There was then nothing with which to compare the results as they were obtained. Basing an estimate on the results of the running test, that for the standing test should probably not be less than three pounds per minute.

These considerations emphasize the undesirability of attaching importance to the comparative showing made by the different coverings during the standing test. The collective showing is, however of more importance.

89. Running Tests and Results.—A complete summary of the facts derived from the running tests is presented as Table XLVIII. The following paragraphs concern such items in the table as require explanation:

"Duration of Test" (Item 2). The general process followed throughout each of the running tests has already been described. As has been stated, every such test included some time during which the engine was at rest. The tests were started before the train was put in motion. Accidental stops operated to increase the duration of the test, and the ten minutes at Maquoketa, during which a balance of the weighing-tank was obtained, are included in the recorded duration of the running test. Finally, at the close, the train came to rest several minutes before it was practicable to end the test. The values, therefore, given as Duration of Test, represent the whole time between the initial and final balancing of the weighing-tank; it is the period for which the record of condensed steam was obtained.

"Actual Running Time" (Item 3). Under this head appears the number of minutes the engine was actually in motion, and under the next head, "Actual Standing Time" (Item 4), is given the difference between the Duration of Test and the Actual Running Time. It will appear further on that the radiation losses, as observed for a whole

TABLE XLVIII.

RUNNING TESTS.

1	Designation of test		Bare A. Boiler.	B.	C.	D_1.	D_2.	E.	F_1.	F_2.	G.
2	Duration of test, min. Time between initial and final balance.	a. 1st Quarter	106.2	86.5	88.7	86.0	88.7	86.1	120.2	85.6	84.5
		b. 2d "	81.0	80.5	76.7	78.4	80.5	77.8	75.6	81.0	79.2
		c. 3d "	75.6	78.8	77.4	73.1	77.8	75.7	80.0	76.6	74.5
		d. 4th "	96.2	112.8	95.4	89.3	97.9	105.5	93.5	103.0	108.2
		e. Total for test	359.0	358.6	338.2	326.8	344.9	345.1	369.3	346.2	346.4
3	Actual running time, min.	a. 1st Quarter	95.0	81.5	82.9	81.5	83.1	81.6	98.5	81.0	81.1
		b. 2d "	70.0	68.3	67.3	68.6	69.0	67.8	69.1	68.6	69.1
		c. 3d "	73.5	73.0	72.7	71.2	73.3	72.5	68.7	73.5	71.2
		d. 4th "	84.7	90.5	83.4	84.3	80.2	86.9	82.5	84.9	79.7
		e. Total for test	323.2	313.3	306.3	305.6	305.6	308.8	318.8	308.0	301.1
4	Actual standing time, min.	a. 1st Quarter	11.2	5.0	5.8	4.5	5.6	4.5	21.7	4.6	3.4
		b. 2d "	11.0	12.2	9.4	9.8	11.5	10.0	6.5	12.4	10.1
		c. 3d "	2.1	5.8	4.7	1.9	4.5	3.2	11.3	3.1	3.3
		d. 4th "	11.5	22.3	12.0	5.0	17.7	18.6	11.0	18.1	28.5
		e. Total for test	35.8	45.3	31.9	21.2	39.3	36.3	50.5	38.2	45.3
5	Average boiler pressure, pounds by gauge.	a. 1st Quarter	146.6	148.2	155.3	150.8	149.9	150.6	149.9	150.6	152.1
		b. 2d "	146.1	150.9	148.3	150.2	148.5	151.3	150.8	152.3	151.8
		c. 3d "	143.2	151.5	153.8	152.2	151.6	151.4	152.3	151.4	151.6
		d. 4th "	147.4	150.1	152.6	150.6	152.0	151.8	150.3	150.1	151.0
		e. Total for test	145.8	150.1	152.6	151.0	150.5	151.3	150.7	151.0	151.6
6	Average atmospheric temperature, F.	a. 1st Quarter	79.8	74.3	81.3	72.8	76.8	81.3	81.5	79.0	78.2
		b. 2d "	85.0	79.3	82.0	74.0	79.2	86.0	84.3	80.0	79.8
		c. 3d "	82.0	85.8	82.0	76.3	79.3	90.7	90.0	81.0	84.6
		d. 4th "	82.0	83.3	81.3	77.0	78.0	90.3	88.2	81.0	83.3
		e. Total for test	81.7	80.3	81.6	75.0	78.2	86.8	85.5	80.3	81.1

TABLE XLVIII—(Continued).

RUNNING TESTS.

7	Direction and force of wind, miles per hour. N	a. 1st Quarter	10 to 12	2	10 to 15	5	3 to 4	5 to 6	5 to 7	3 to 4	2 to 3
		b. 2d "	10 to 12	2	10 to 18	5	3 to 4	10	6 to 10	3 to 4	2 to 3
		c. 3d "	10 to 12	2	10 to 18	5	3 to 4	10	5	2 to 4	3
		d. 4th "	10 to 12	2	10	5	3 to 4	10	Calm	2 to 4	3
8	Condition of sky.		Hazy, sunshine	Hazy,	Clear	Cloudy, variable	Clear	Bright, hazy	Alternately bright and cloudy	Clear	Clear
9	Observed condensation; pounds of water delivered to weighing-barrel.	a. 1st Quarter. . .	1376.9	419.0	545.7	388.4	458.4	397.2	548.8	449.9	467.1
		b. 2d "	1137.8	452.1	413.6	375.2	397.7	416.7	411.9	390.7	418.4
		c. 3d "	975.4	410.7	343.2	393.7	401.9	363.7	378.2	393.7	402.2
		d. 4th "	1168.1	590.5	479.1	472.4	494.3	515.8	437.6	560.3	575.5
		e. Total for test. .	4658.2	1872.3	1781.6	1629.7	1752.3	1693.4	1776.5	1750.6	1863.2
10	Amount to be added to or subtracted from the total observed condensation to convert it into an equivalent amount which would have been observed had the average steam pressure for the test been 150 lbs. and the atmospheric temperature 80°F.		+60.6	0	0	−31	−12.3	+37.3	+32.0	0	+0.9

TABLE XLVIII—(*Continued*).

RUNNING TESTS.

11	Total condensation for test under conditions No. 10	4718.8	1872.3	1781.6	1598.7	1740.0	1730.7	1808.5	1750.6	1864.1
12	Assumed time during which engine was running, minutes	305	305	305	305	305	305	305	305	305
13	Assumed rate of speed, miles per hour	28.3	28.3	28.3	28.3	28.3	28.3	28.3	28.3	28.3
14	Assumed time during which engine was standing. Duration of test minus assumed time engine was running = Item 2e − Item 12	54.0	53.6	33.2	21.8	39.9	40.1	64.3	41.2	41.4
15	Total condensation during running = condensation for test minus the product of the rate of condensation while standing and the assumed standing time	4352.7	1753.2	1668.1	1535.3	1628.3	1589.5	1613.0	1617.9	1738.7
16	Condensation per minute while running 28.3 miles per hour	14.27	5.74	5.47	5.03	5.34	5.21	5.29	5.30	5.70
17	Reduction in condensation per minute while running at a speed of 28.3 miles per hour, resulting from covering applied to 61% of total surface of boiler, pounds	....	8.53	8.80	9.24	8.93	9.06	8.98	8.97	8.57
18	Ratio of heat saved by covering to total heat transmitted from bare boiler	....	0.598	0.617	0.648	0.626	0.635	0.629	0.628	0.601

test, can readily be separated into two parts, one applying to the standing time and the other to the running time. In this manner results are obtained which apply wholly to the conditions of running, the effect of all stops being eliminated.

"Corrections for Variations in Time of Running and in Speed" (Items 12 to 14). Attention has already been called to the fact that the running tests are actually made up of intervals during which the engine was in motion, and of other intervals during which it was at rest. This being so, and the time of running and the time of standing being known, it is easy to divide the observed condensation into two parts, one of which shall represent that which resulted during the period when the engine was standing, and the other that which resulted during the time the engine was in motion. Thus, the duration (Item 2) of Test B is given as 358.6 minutes, the actual running time (Item 3) as 313.3 minutes, and the standing time (Item 4) consequently as 45.3 minutes, and the total condensation (Item 11) is 1872.3 pounds. The rate of condensation while standing was shown to be 2.63 pounds per minute (see results from Standing Test, Table XLVII)., so that during the 45.3 minutes the engine was standing 119.1 pounds were condensed. The condensation while running, therefore, would be the total condensation minus the condensation while standing, that is:

$$1872.3 \text{ pounds} - 119.1 \text{ pounds} = 1753.2 \text{ pounds}.$$

A determination in this general form has been made for all tests in order that the rate of condensation during the running time may be shown. Instead, however, of taking the actual running time and the actual standing time, there has been employed an assumed running time (Item 12), which is the same for all tests, and which is very near the actual time of all tests. The adoption of this assumed running time is justified from the following considerations: If, for a given test, the running time is slightly longer than for another test with which it is to be compared, it is evident that its rate of speed was lower and that the air-currents, as a consequence would, other things being equal be less severe in their effect upon the engine. There would, therefore, be injustice in determining the rate of condensation per minute in the two cases for purposes of comparison by dividing the total condensation by the observed time. If, however, the total condensation observed during running is, in each case, divided *by the same time*, the result in each case carries with it its own correction for variations

in speed of running. This is the process which has been followed in making up the statement of results. At the same time full and complete data are given which will permit a comparison to be made on the direct basis, if any are disposed to mistrust the assumptions already stated. The conclusion of this matter is represented by the "Condensation per Minute while Running" (Item 16). This factor represents, therefore, the observed condensation as corrected for variations in steam pressure, for variations in atmospheric temperature, and for variations in speed of running. It is probably as perfect a basis upon which to compare the merits of the various coverings tested as can be supplied by the information obtained during the tests. It is evident that since all condensation must have resulted from heat radiated, the smaller the amount of condensation the more efficient the covering.

90. Conditions Affecting Results for which no Corrections have been Applied are to be found in the varying thicknesses of covering experimented upon, and in the varying velocity and varying direction of the wind.

It may be concluded from results obtained that corrections for varying thicknesses, were it practicable to derive them, would be extremely small in value. Nevertheless, the effect of differences in the thickness of the several coverings is not in fact negligible, if results are to be directly compared.

The cooling effect of the wind will vary with its velocity and direction. When its direction is the same as that in which the train travels, its effect upon the locomotive is similar to that which would be produced if the locomotive were moving through still air at reduced speed. As each test involved a round trip over a given line of track, the direction of train motion was reversed in the middle of the test, thus reversing the effect of the wind, the average effect for the whole test remaining practically the same as though the whole movement of the locomotive had been in still air. This, of course, is strictly true only when the direction of the wind is in line with the track, but the argument has force in connection with all tests made. The tabulated statement of condensation by quarters (Item 9), compared with wind diagrams (Item 7), is instructive on this point. Thus, taking the second and third quarters, from Maquoketa to Anamosa, and from Anamosa to Maquoketa respectively, Test B shows a constant direction of wind and a velocity of about two miles. From Maquoketa to Anamosa, *against* the wind, the condensation is 452.1

pounds; from Anamosa to Maquoketa, *with* the wind, the condensation is 410.7 pounds, a difference of 41.4 pounds. There can be but little doubt that the average of the two values will be very close to the result which would have been obtained had the engine been running in still air once over the line, at the speed which prevailed during the test.

Comparisons of this kind should be made with care. For example, from considerations just presented, it would appear that the condensation for the last quarter should be less than for the first quarter, whereas, the uncorrected data, with which we are now concerned, show it to be greater. The explanation is to be found in the fact that the return trip involved detentions, which made the time returning on the fourth quarter nearly half an hour longer than the outward time of the first quarter. The corrections for such irregularities have been applied to the results of the whole tests only, and not to the separate quarters.

Returning, again, to a consideration of wind effects it is to be noted that changing the direction of train motion does not compensate for the effect of side-winds. For this reason, such winds, even when light, doubtless have a more serious effect in impairing the comparative value of the results than stronger winds which move along the line of the track.

Again, changes either in the direction or velocity of wind, during the progress of a test, constitute a source of serious disturbance. In Test D_1 the wind moved with the engine during the first quarter, and, later in the test, changed so as to move obliquely with the engine during the return trip, with the result that the condensation for the whole test is probably somewhat less than it would have been had the direction remained unchanged. A similar change of wind, Test E, was against the engine, its effect being to give a greater amount of condensation than would have been obtained had the wind remained unchanged.

91. A Summary of Results.—The observed and calculated results are given in detail in Tables XLVII. and XLVIII. A summary of these results is here given as Table XLIX.

The values as given have been reduced to a common basis with reference to steam pressure, atmospheric temperature, and running speed, and, so far as these factors are concerned, are comparable. They have not been corrected for variations in thickness of covering, which in all cases were slight, or for variations in the velocity and direction of the wind.

TABLE XLIX.

POUNDS OF STEAM CONDENSATION PER MINUTE.

	A (Bare Boiler.)	B	C	D_1	D_2	E	F_1	F_2	G
Standing test. . .	6.78	2.63	3.42	2.91	2.80	3.52	3.04	3.22	3.03
Running test. . . Speed, 28.3 miles.	14.27	5.74	5.47	5.03	5.34	5.21	5.29	5.30	5.70

92. **Efficiency of Coverings.**—The percentage of the heat transmitted from the bare boiler, which is saved by any covering, may be obtained by subtracting the amount of condensation for the covering in question from the condensation for the bare boiler, and by dividing one hundred times this difference by the condensation for the bare boiler. The result expresses the efficiency of the covering. Values thus obtained are given in Table L.

TABLE L.

EFFICIENCY OF COVERINGS AS DISCLOSED BY RUNNING TESTS (PER CENT).

B.	59.8
C.	61.7
D_1.	64.8
D_2.	62.6
E.	63.5
F_1.	62.9
F_2.	62.8
G.	60.1

The results of this table are corrected for variations in steam pressure, atmospheric temperature, and speed, but not for variations in weather and wind conditions, or for variations in thickness of covering. The conclusion to be drawn from them, stated in very general terms, is that any of the coverings tested can be relied upon to save from 60 to 64 per cent of all the heat which would radiate from the boiler were it not covered at all. A fairly representative result may be stated as 62.3 per cent.

The fact that the results obtained from the several coverings are so nearly alike can hardly fail to occasion surprise. Had thin layers of the material tested been subjected to carefully planned lab oratory tests, the results would doubtless have differed more widely

but it must be expected that the value of such difference will diminish as the specimens experimented upon are increased in thickness. A material which is rather an indifferent non-conductor will serve to prevent the passage of heat, if applied in sufficient thickness. While, therefore, the coverings tested were of normal thickness, it would seem that this thickness is sufficient to reduce to a negligible amount the effect of the superior non-conducting properties which the material of one covering may have possessed over others.

The results show that the covering of 61 per cent of the exterior surface of the experimental boiler saves 62.3 per cent of all the heat radiated from the same boiler under similar circumstances when bare. It does not, however, follow from this statement that if 100 per cent of the exposed surface of the boiler were covered, 102 per cent of the heat lost from the bare boiler would be saved. Such a conclusion must obviously be absurd, though a hasty consideration of the facts presented might seem to justify it. The fact, as first stated, however, proves that there is a vast difference in the character of the exposure to which different portions of the boiler are subjected. While only 61 per cent of the surface of the boiler was covered, the protection was evidently applied where it was most needed. The percentage of the *total exposure* guarded against was greater than the percentage of surface covered. For this reason increasing the covered area by 10 per cent cannot be depended upon as a means of reducing radiation losses by a like amount. It will reduce loss, but the amount of the reduction may be very much less than 10 per cent. It is for this reason, also, that all comparisons in this report have been based upon the boiler as a whole. The radiation is stated in terms of pounds of steam condensed per minute for the boiler experimented upon, rather than as pound per minute per square foot of exposed surface. The latter unit would be a more general unit, but its use in interpreting the data under consideration would be misleading.

93. **Radiation and its Power and Coal Equivalent.** — Assuming that a locomotive will develop a horse-power by a consumption of twenty-six pounds of steam per hour, and assuming that the steam thus consumed must be generated from water at 80° F., the radiation losses already given may be expressed in terms of power losses of equal value. The practical effect of these assumptions is to define a horse-power as equal to the condensation under the conditions of the tests of thirty-four pounds of steam per hour, the steam having a pressure of 150 pounds and the water the temperature due to this

pressure. Upon this basis the following results are obtained. They apply only to the boiler tested.

TABLE LI.

POWER LOST BY RADIATION.

	Horse-power Equivalent to Radiation Losses.
Bare Boiler:	
Locomotive at rest under conditions of test.	12
Locomotive running 28.3 miles per hour and otherwise under conditions of test.	25
Covered Boiler:	
Locomotive at rest under conditions of test.	4.5
Locomotive running 28.3 miles per hour and otherwise under conditions of test.	9.3

Under ordinary conditions of operation four pounds of coal are consumed per horse-power hour, hence the coal equivalent to the radiation per hour may be found by multiplying the values of Table LI. by 4. Thus:

TABLE LII.

COAL REQUIRED TO MAINTAIN RADIATION LOSSES.

Bare Boiler:	
Locomotive at rest under conditions of test.	48 lbs.
Locomotive running 28.3 miles per hour and otherwise under conditions of test.	100 lbs.
Covered Boiler:	
Locomotive at rest under conditions of test.	18 lbs.
Locomotive running 28.3 miles per hour and otherwise under conditions of test.	37 lbs.

94. The Effect of Conditions other than those which Prevailed during the Tests.—The fact should be emphasized that the results thus far given are those derived from the actual experiments. These involved a boiler of moderate size, carrying steam pressure which is now regarded as low, and were conducted in the month of August. It should be noted, also, that the running tests involved a speed of less than thirty miles per hour. It is evident that other conditions, quite common to actual service, would operate to greatly increase the radiation losses described. The effect of changes in some of these conditions will next be considered.

The effect on radiation of changes in speed has long been an open question. It has been argued that a boiler perfectly covered would

be, to a very great extent, unaffected by surrounding air-currents, and hence that its radiation losses would not be materially greater when the locomotive is at speed than when standing. But those who appreciate the intensity of the cooling currents, which circulate about a locomotive when at speed, have been slow to accept such a view, and the tests under consideration confirm their position. They give a measure of the radiation losses, both when the locomotive is at rest and when moving at a uniform speed of 28.3 miles an hour. While these

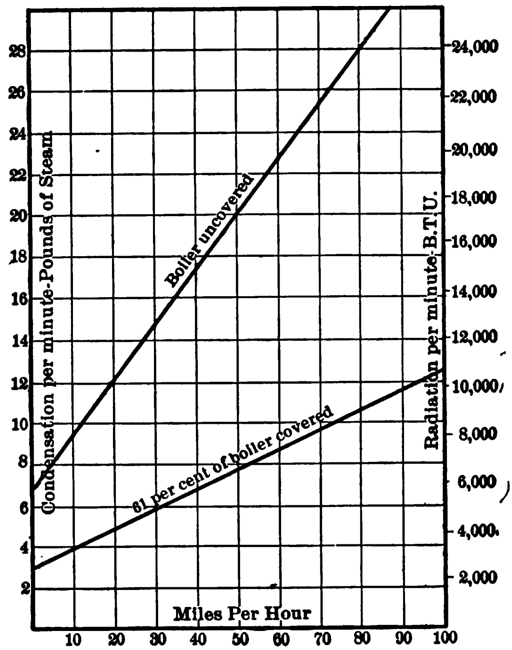


FIG. 102.—Effect of Speed on Radiation Loss.

points are not sufficient to establish with accuracy the complete relationship of radiation and speed, an estimate of real value may be based upon them. Such an estimate is presented in the form of a diagram, Fig. 102. The diagram shows that the bare boiler when at rest radiates sufficient heat to condense 6.8 pounds of steam, at 150 pounds pressure per minute, which amount is increased to twenty-eight pounds when the same boiler is driven at a speed of eighty miles an hour. Similar values for the covered boiler are 3.0 pounds and 10.6 pounds respectively.

Changes in atmospheric temperature would of necessity affect results. Those recorded were obtained in midsummer and all have been corrected for an atmospheric temperature of 80° F. For each 10° reduction in atmospheric temperature below 80° the radiation may be expected to increase 3.5 per cent. For zero degrees temperature the radiation losses recorded in this report should be increased by about 28 per cent. For example, if, when the atmospheric temperature is 80°, the conditions are such as result in the condensation of five pounds of steam per minute, when the atmospheric temperature is 0° the condensation will be

$$5+5(.035\times 80)=5+1.4=6.4.$$

From this it appears that very low temperatures are attended by radiation losses of considerable magnitude.

Changes in steam pressure also would affect results. The experiments were conducted under a boiler pressure of 150 pounds by gauge. With an increase of pressure the boiler temperature will become higher, and the radiation losses will, as a consequence, be augmented. Changes arising from this source, however, are not great. For each ten-pound increase of pressure above the limit of 150 pounds the radiation may be expected to increase by about 1.6 per cent, but this will not apply for pressures much above 200 pounds. A pressure of 200 pounds will involve losses by radiation which are 8 per cent greater than those making up the record of this report.

From these considerations it can be shown that with the boiler bare and the locomotive running at eighty miles an hour, under a steam pressure of 200 pounds, with the atmospheric temperature 0°, the loss by radiation would be the equivalent of sixty-seven horse-power, while a covered boiler, running under the same conditions of speed, pressure, and atmospheric temperature, would still be subject to a loss of twenty-five horse-power. As a locomotive similar to that tested may be expected to deliver a maximum of 600 horse-power, it is evident that under the extreme conditions just assumed at least 10 per cent of the total power of the engine would be lost in radiation. This is for an uncovered boiler. An application of any of the coverings tested would reduce this maximum loss to about 4 per cent.

Finally, it should be remembered that the boiler tested was one of moderate size. Many boilers are now running which present an exposed area which is at least 50 per cent greater than that presented by the boiler under test, and it should be evident that the losses from

such large boilers will be greater than those disclosed by the tests under consideration. For boilers of the same general type the loss will probably be proportional to the exposed surface.

95. Conclusions concerning the extent of losses by radiation may be stated as follows:

1. The amount of heat radiated from the boiler of a locomotive in motion is affected by many different conditions, as, for example, by the speed of the locomotive, the temperature of the atmosphere, the direction and velocity of the wind, the steam pressure, and by the pro portion of the whole surface of the boiler which is covered.

2. The application of a suitable covering to 61 per cent of the exposed surface of the boiler has resulted in a saving of 62 per cent of the heat radiated from the uncovered boiler. This does not imply that if the whole surface were covered the radiation would be reduced to zero, but rather that the covered portions are those which are most exposed.

3. The radiation from a given boiler is approximately twice as great when the locomotive is running thirty miles an hour as when it is at rest.

4. The loss from a bare boiler of a locomotive running 30 miles an hour is from 4 to 10 per cent of the maximum capacity of the boiler. In summer it will not much exceed the lower limit, in winter it may frequently approach the maximum. An increase of speed to 60 miles will nearly double the loss.

5. The loss from a boiler of a locomotive running 30 miles an hour having a considerable portion of its surface covered in accord with good practice, is from less than 1 to not more than 5 per cent of its maximum power. In summer it will not greatly exceed the lower limit; in winter it may frequently approach the maximum. An increase of speed to 60 miles an hour will augment the loss to 1.3 that which occurs at 30 miles.

CHAPTER XI.

THE FRONT END.

96. Definitions.—For the purpose of this discussion the term "front end" refers to all that portion of a locomotive boiler which is beyond the front tube-sheet. It includes the extending shell of the boiler which forms the smoke-box, and in general all mechanism which is therein contained, such as steam- and exhaust-pipes, netting, diaphragm and draft-pipes. It also includes the stack.

The front end as thus defined is to be regarded as an apparatus for doing work, receiving energy from a source of power, and delivering a portion thereof in the form of a specific result. The source of power is the exhaust steam from the cylinders, and the useful work accomplished is represented by the volumes of furnace gases which are delivered against the difference of pressure existing between the smoke-box and the atmosphere. That the power of the jet may be sufficient, it is necessary that the engines of the locomotive exhaust against back pressure. The presence of the back pressure tends to lower the cylinder performance, and it is for this reason that designers of front ends have sought to secure the required draft action in return for the least possible back pressure. In other words, the effort has been to increase the ratio of draft to back pressure, which ratio has been defined as the efficiency of the front end.

97. Draft and its Distribution.—The office of the front end is to draw atmospheric air into the ash-pan, thence through the grate and fire, to draw the furnace gases through the tubes of the boiler, thence under the diaphragm and into the front end, and to force them out into the atmosphere. In order that this movement may take place, a pressure less than that of the atmosphere is maintained in the smoke-box, so that when the locomotive is working there is a constant flow from the atmosphere along the course named and back to the atmosphere again. The difference in pressure between the atmosphere and the

smoke-box is spoken of as the draft, and, under normal conditions of running, is represented by from 4″ to 10″ of water.

It has often been asked whether the draft, as observed from gauges attached to the front end, is in any way affected by changing the point of application of the gauge, a question which is not unreasonable in view of the intense activity which characterizes the circulation of gases through the front end. In undertaking a study of the draft action at Purdue, it was early determined to settle this point. Elaborate apparatus was prepared which would serve in giving simultaneous observations of the pressure at several different points within the front end, and which by adjustments which could be quickly made, could be arranged to apply to a new series of points. By the use of this apparatus it was found possible, in a brief interval of time, to explore thoroughly the condition of pressure within the front end, and to make

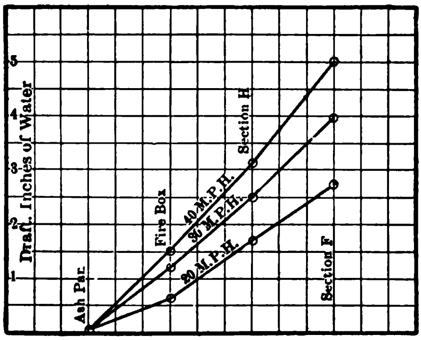


FIG. 103.—Distribution of Draft.

of record the pressures at eighty-four different points, the time required being so short as to leave no question as to the constancy of the running conditions. Many different sets of observations thus obtained justified the conclusion that, so far as the space in front of the diaphragm is concerned, draft-gauges attached at any points will all show identical readings.*

The distribution of draft throughout the apparatus intervening between the front end and the ash-pan is a matter of more than ordinary interest. Its value, as determined when the locomotive was operating under three different rates of power, is shown by Fig. 103. Calling

* For a detailed description of this work see "American Engineer," May, 1902. p. 131.

the ash-pan pressure zero and referring to the 40-mile curve of this diagram, the draft in the fire-box, as measured by a gauge attached to a hollow stay-bolt, is 1.5″ of water, between the front tube-sheet and the diaphragm (section H) it is 3″, and between the diaphragm and the base of the stack (section F) it is 5″. The last value is that which is usually measured and referred to as the "draft." The percentage of the total draft absorbed by various portions of the system is shown by Table LIII.

TABLE LIII.

PERCENTAGE OF TOTAL DRAFT REQUIRED.

Speed, Miles per Hour.	To Draw Air into Fire-box.	To Draw Smoke and Gases through Tubes.	To Draw Smoke and Gases under Diaphragm.
20	22.6	41.1	36.3
30	30.1	33.6	36.3
40	30.4	32.0	37.6

It will be seen that the relative resistance of different portions of the system varies but little under changes in speed. More than one-third of the total draft value is absorbed in drawing the gases past the diaphragm, approximately another third is absorbed in the passage of the tubes, and the remainder is available for drawing air from the ash-pan through the grate and fuel into the fire-box. It is really this latter value only which promotes combustion. Under present conditions of operation, it is thought necessary as a matter of practical importance to have the opening under the diaphragm ocmparatively small, in order that the velocity of the gases may be sufficiently high in passing to clear the front end of cinders. The results show that the cost of this self-clearing action, as measured in the draft absorbed, is so great as to suggest the possibility of a material improvement in design at this point. If the diaphragm could be abandoned, or if the area under it could be materially increased without introducing new difficulties, the efficiency of the front end could be greatly improved.

It is often assumed that a locomotive having a wide fire-box and large grate can be operated with a larger exhaust-tip than one having a narrow fire-box and smaller grate and that since the larger grate permits a given amount of fuel to be burned at a lower rate of combustion, the work which the steam-jet has to do is diminished. Experience on the road, on the other hand, has shown that differences in grate area do not necessarily involve, or ordinarily permit, changes in the draft

appliances. The reason for this is to be found in the fact that, whether the grate be large or small, two-thirds of the difference in pressure between the front end and the atmosphere is absorbed in moving the air and gases through the tubes and under the diaphragm. The volume of air and gases to be moved is not materially changed by changing the area of the grate, and while the difference in pressure between the atmosphere and the fire-box will be less with the large grate than with the small one, this difference is insignificant in amount when compared with the total draft action which is required to stimulate the movement through the remaining portion of the systems.

98. The Action of the Exhaust-jet.—A study of the form and action of the jet of exhaust steam in the front end of a locomotive was undertaken at the Purdue Laboratory, in coöperation with a committee of the Master Mechanics' Association.* Prior to this time it had been common to assume that the action of the exhaust-jet was similar to that of a pump, that the exhaust from each cylinder-end supplies a ball of steam which fills the stack very much as the piston of a pump fills its cylinder, and which, by virtue of its own momentum, pushes before it a certain volume of the smoke-box gases until it passes out of the top of the stack. In obedience to this theory, experimenters had sought to so design their apparatus that the spread of the jet would be sufficient to fill the stack at its base. A knowledge of the form of the jet was therefore regarded as of prime importance. With this understanding of the general problem the Purdue laboratory entered upon its work.

The apparatus employed is shown in part in connection with Fig. 104, which is a drawing to scale, showing a cross-section through the center of the stack of the front end experimented on. In line with the center of the stack and exhaust-pipe, cast-iron sleeves were fastened to the outside of the smoke-box. Through these were fitted pipes 1, 2, 3, 4, and 5, arranged to slide in and out across the smoke-box, and having their inner ends turned down to a fine tip with sharp edges, and the edges bounding the orifices. The body of each pipe was graduated to tenths of inches, the scale reading from a reference-mark fixed to the sleeve on the outside of the smoke-box. If the zero of the scale were brought under the reference-mark, the inner tip of the pipe would be directly under the center of the stack; that is, directly over

* For a full account of the important work accomplished by this committee, see "Proceedings of the American Railway Master Mechanics' Association," Vol. XXIX., 1896.

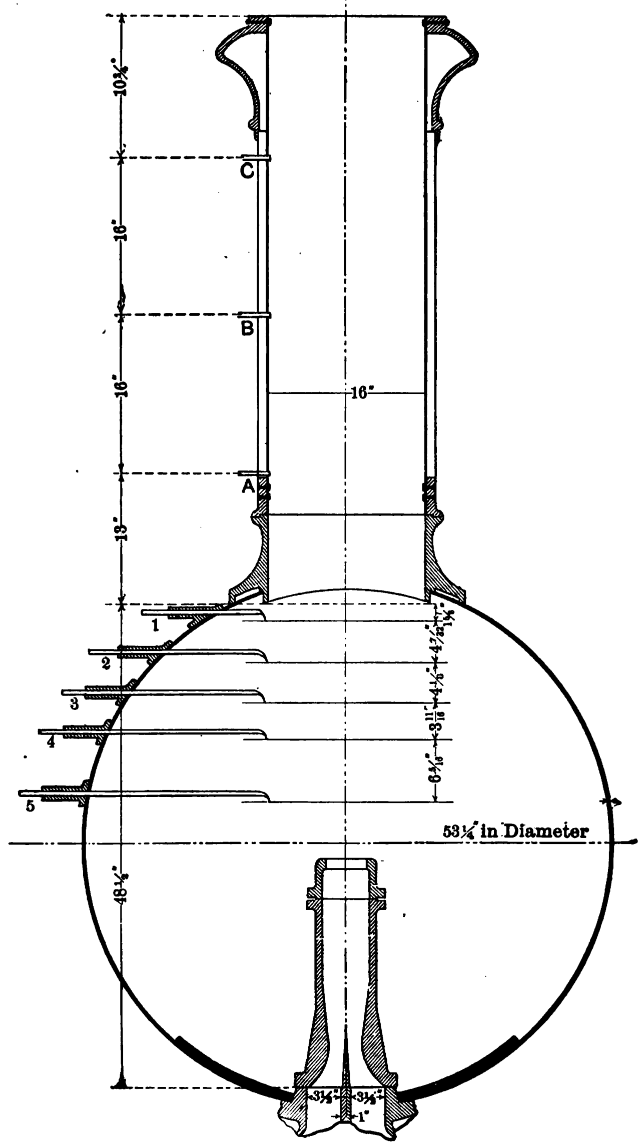


FIG. 104.

the center of the exhaust-pipe. It is obvious that if the tip of any pipe be surrounded by the jet of exhaust steam, the velocity of the latter will tend to carry steam through the pipe and to discharge it into the atmosphere outside of the smoke-box. It can be shown also that the force which the steam would exert in its effort to pass the pipe is a function of its velocity; hence, by observing the force or pressure, the velocity may be calculated. Pressures were observed by having the outer ends of each sliding-pipe connected by rubber tubing with one leg of a manometer or U shaped glass tube, which was fastened to the wall of the laboratory. These U tubes were partially filled with mercury, the displacement of which gave the pressure transmitted through the tube. If, for example, the tip of any particular pipe were in the jet of steam, its manometer would show pressure greater than that of the atmosphere; if it were withdrawn from the jet its manometer would indicate a pressure less than that of the atmosphere.*

Besides these sliding-pipes in the smoke-box, the stack was fitted with three pipes which had plain ends projecting beyond the inner wall about a quarter of an inch. These side orifices were each connected with a manometer. They served to show the extent of pressure or vacuum existing within the stack at points where they were attached. Their exact location is shown by the dimensions in Fig. 104.

The normal draft was measured by two different manometers connected with the front end, and was permanently registered by a Bristol recording-gauge. Indicators were used to show the steam distribution in the engine cylinders, and a special indicator fitted with a light spring gave a fine record of the back-pressure line. This pressure was also recorded by a Bristol gauge. A Boyer speed-recorder served as a means for maintaining constant speed conditions.

Observations were made as follows: Desired conditions of speed and steam pressure having been obtained, the adjustable pipes (1, 2, 3, 4, and 5), Fig. 104, were withdrawn from the smoke-box sufficiently to bring them entirely clear of the exhaust-jet, usually to a distance of 4½ inches from center of jet. Then, upon signal, all manometer gauges were read and all other observations taken, the readings being taken simultaneously. Each sliding-pipe was then moved inward a tenth of an inch and readings repeated, after which they were moved

* See "A Glimpse of the Exhaust-pipe," Proceedings of the Western Railway Club for October, 1895.

another tenth, and so on until the tip of all the pipes reached the center of the exhaust-jet. The readings thus obtained from the pipes 1, 2, 3, 4, and 5, Fig. 104, were entered upon a half-sized drawing representing a portion of the cross-section of the smoke-box, the position of each entry showing the exact location to which the numerical readings applied. Upon the diagram of pressures thus obtained lines were drawn through points for which the pressure was zero. These lines were assumed to represent the border of the steam-jet. Reduced in scale, they are the full lines given in Figs. 107 to 118, the full significance of which is to be hereafter considered. Next, lines were constructed by connecting points showing a pressure of 0.5 inch of mercury, and still other lines by connecting points representing a pressure of 1.5 inches of mercury; and, finally, in some cases the curves corresponding with 2.5 inches of mercury were located. These lines of equal pressure are those which appear in Figs. 107 to 118, but the numbers given in connection therewith represent the velocity in feet per second, as calculated from the pressure already noted. The curves are, in fact, equal velocity curves as well as curves of equal pressure.

Confining the discussion for the present to matters affecting the action of the jet, it may be said with certainty that the exhaust-jet acts upon the smoke-box gases (1) to induce motion in those portions which immediately surround it, and (2) to enfold and entrain the gases which are thus made to mingle with the substance of the jet itself.

The induced action, which, for the jets experimented upon, is by far the most important, may be illustrated by means of Fig. 105. The arrows in this figure represent, approximately, the direction of the currents surrounding the jets. It will be seen that the smoke-box gases tend to move toward the jet, and not toward the base of the stacks, at which point they are to leave the smoke-box. That is, the jet, by virtue of its high velocity and by its contact with surrounding gases, gives motion to particles close about it, and these, moving on with the jet, make room for other particles which are farther away. As the enveloping shell of gas approaches the top of the stack its velocity decreases and it becomes thinner and thinner, all as shown by Fig. 105. All parts of the jets require gases to work upon, the upper as well as the lower part. Gauges attached to the side of the stack show a vacuum, because the gases needed for the upper portion of the jet can reach it only by coming in around the jet lower down. In other words, the action of the upper part of the jet induces a vacuum in the lower part

of the stack, just as the action of the jet as a whole induces a vacuum in the smoke-box. It will be shown later that as the amount of work

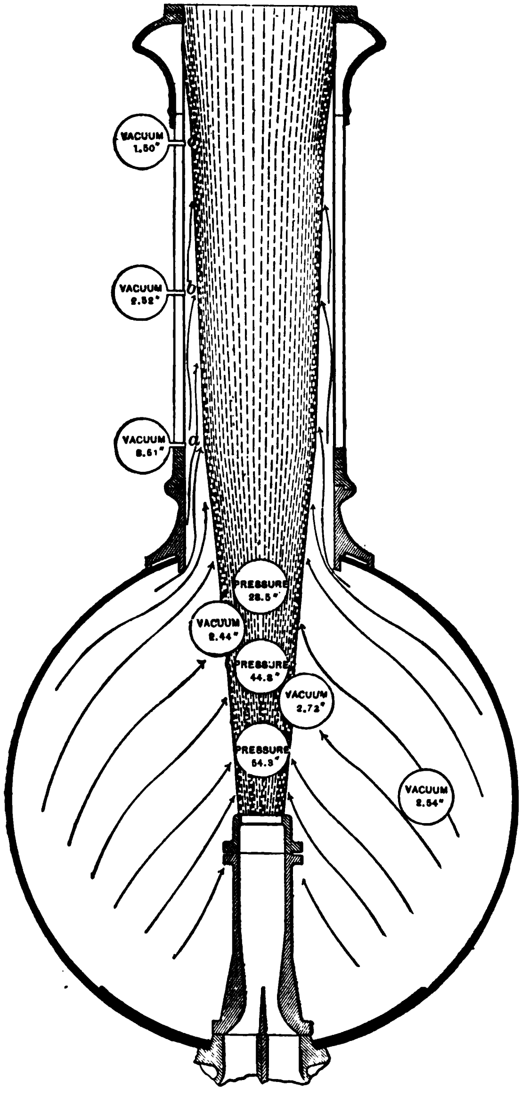


FIG. 105.

to be done by the exhaust is increased the jet becomes smaller, thus making room for larger volumes of gas to pass between it and the

stack; the velocity, both of the jet and of the induced currents, increasing. It is of interest in this connection also to note that of the gauges attached to the side of the stack (Fig. 105), *a*, which is 13 inches above the base, always gave about one and one-half (1.5) times the reduction of pressure which was recorded by the gauge attached to the smoke-box, and that the second gauge, which is 10 inches from the top of the stack, gave approximately six-tenths (0.6) of the value recorded in the smoke-box.

That there is some intermingling of the smoke-box gases with the steam of the jet is made evident by the appearance of the combined stream as it issues from the top of the stack. The manner in which this intermingling takes place will be seen from the following considerations:

Any stream flowing from a nozzle through a resisting medium will have a higher velocity at its center than at its circumference or sides, that is, the particles at the center of the jet move at a higher velocity than those on the outside, the latter being held back by contact with the surrounding gas. The result of the different velocities in the same stream is a wave motion of the individual particles of which the stream is composed. Thus, the path of any one of these particles may be shown by Fig. 106, B, but the exact form and frequency of the loops

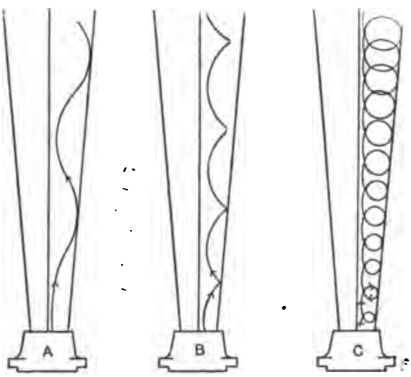


FIG. 106.

will depend upon the relation between these differences in the velocity of particles in different portions of the stream and the actual mean velocity of the jet. If the velocity of the jet is high, and differences for different portions of the cross-section are not great, the loop may disappear, the path appearing as shown by Fig. 106, A. With a still

higher mean velocity and a smaller difference the loops would approach the form shown by Fig. 106, C. The exhaust-jet appears to take this latter form. Measurements to determine its velocity show that particles in the center move much more rapidly than those near the outside, and other measurements to determine the form of the jet as a whole define a boundary which is neither a straight line nor a regular curve, but which agrees closely with the form given by Fig. 106, C. All this shows that the jet is stepped off in nodes, which under given conditions remain fixed in position. This conclusion, based upon measurements of the jet, is confirmed by the appearance of the jet as seen in an engine running with the front end open. The jet, when thus viewed, exhibits one or more bright spots which remain in a fixed position. It is through the wave-like action of the particles making up the steam-jet that the surrounding gases are intermixed with the jet.

It is clear that any design of nozzle which will serve to subdivide the stream, or to spread it so as to increase its cross-section, will assist the jet in its effort to entrain the gases, but it is not clear that there is any gain to be realized in such a result. It is possible that as the mixing action is increased the induced action may be diminished, and that the sum total of the effect produced may remain nearly constant. The work which has thus far been done is not conclusive on this point, but the evidence tends to show that the more compact and dense the jet the higher its efficiency. It is certainly clear that for the jets experimented upon the mixing action is hardly more than incidental to the induced action, the latter constituting the influence through which the work of the jet is chiefly accomplished.

The similarity existing between the action within the front end of a locomotive and that within an injector will at once suggest itself, but the resemblance is not perfect. In an injector the jet of steam is almost immediately and completely condensed. It loses its identity by intermixing with the water to which it imparts its energy. The mixture is complete before the water-orifice, through which the combined stream must pass, is reached. In the front end of a locomotive, on the other hand, the mixing of the jet with the stream of gas upon which it acts proceeds but slowly, and with the apparatus now in use it is doubtful if the process is ever completely accomplished.

99. Form and Character of the Jet.—In Figs. 107 to 118 the steam-jet is represented by a series of lines. The full or outside lines are assumed to represent the boundary of the jet. They do, in fact, pass through points for which the velocity of the steam and gases is so low

that their impress upon the bent tubes is only sufficient to balance the pressure of the atmosphere. Each dotted line inside of those already referred to passes through a series of points where the velocities are equal, the value of which in terms of feet per second is given. A comparison of the form and location of similar lines in different diagrams will show the effect produced by the different combinations of mechanism employed.

100. The Jet as Affected by Changes in Speed of the Locomotive.—The three jets shown by Figs. 107 to 109, inclusive, were obtained

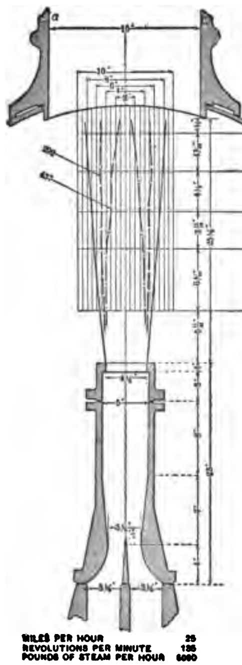


FIG. 107.

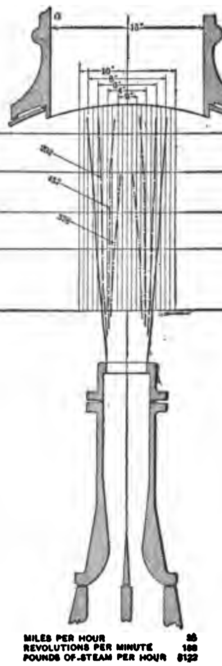


FIG. 108.

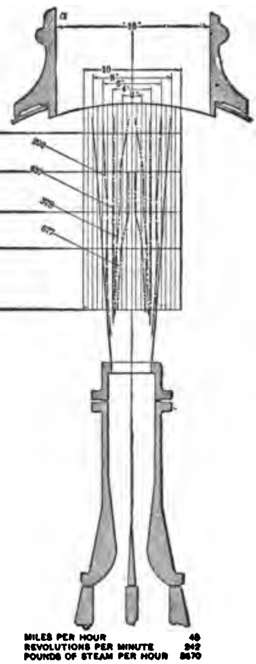


FIG. 109.

under similar conditions, except that for Fig. 107 the speed was 25 miles per hour; for Fig. 108, 35 miles; and for Fig. 109, 45 miles. Each increment of speed results also in a larger volume of steam delivered, and, consequently, a greater reduction of pressure. The diagrams show that the rapidity of the exhaust impulse does not greatly affect the character of the interior of the jet, also that the spread of the jet diminishes as the speed is increased; or, in other words, as the volume of steam passing the nozzle becomes greater, the increased

velocity of both the jet of steam and the body of gas immediately surrounding it, together with the reaction of the latter, probably constitute the causes which operate to narrow the jet. The velocity curves, which in Fig. 108 are pretty evenly distributed throughout the body of the jet, are in Fig. 109 crowded together, giving evidence in the latter case of a very dense and powerful jet. The results show that the more rapidly moving, though smaller, jet (Fig. 109) is more effective in producing draft than that which is shown by Fig. 107.

101. **The Effect upon the Jet of Changes in the Height of the Bridge.**—An essential feature of the single exhaust-pipe is the so-called bridge, which maintains a separate steam-passage for each cylinder for a portion of the length of the pipe. It is above the top of this bridge only that the steam exhausted from both sides of the locomotive intermingles. The purpose of the bridge is a twofold one: it prevents the exhaust of one side from blowing through into the exhaust-passage of the other side, and when properly designed it is the means of making that side from which the strongest stream is passing assist by induction the exhaust from the other side. To determine the best height of bridge exhaust-pipes of four different designs were tested by the Master Mechanics' Committee, all being the same except in the height of the bridge and in the proportions of those parts adjacent thereto. Two nozzles representing the extreme conditions with reference to this detail were employed in experiments designed to determine the character of the jet, with results which are to be seen by comparing Figs. 107 to 109 with Figs. 110 to 112. Figs. 107 and 110 represent identical conditions, except as to the height of the bridge, as do also Figs. 108 and 111, and Figs. 109 and 112. An examination of the velocity curves of these figures will show that for similar conditions of running the jet for the two groups is nearly identical. It will be of interest to add that, basing their conclusions upon efficiency tests made at Chicago, the committee concluded that, whenever the bridge was less than 12″, some loss of efficiency resulted, though such loss was not great even when the height of the bridge was reduced to 5″. They recommend that the bridge be not less than 12″. They recommend, also, that where long exhaust-pipes must be employed, the increased length should always be secured by extending that portion of the pipe which is above the bridge. This recommendation in effect contemplates a constant height of bridge for all lengths of pipes.

To the discussion with reference to the bridge it will be of interest to add some reference to the "choke."

That part of the steam-pipe between the bridge and the outside wall, where the contraction of the passage is greatest, has generally been referred to as the choke. An early committee had recommended that the area of the choke be 80 per cent of that of the nozzle, and the committee of 1896 did a large amount of work upon pipes having this proportion. Not satisfied, however, with the validity of the conclusions, the later committee extended its research to involve a large range of proportions, with results sustaining the con-

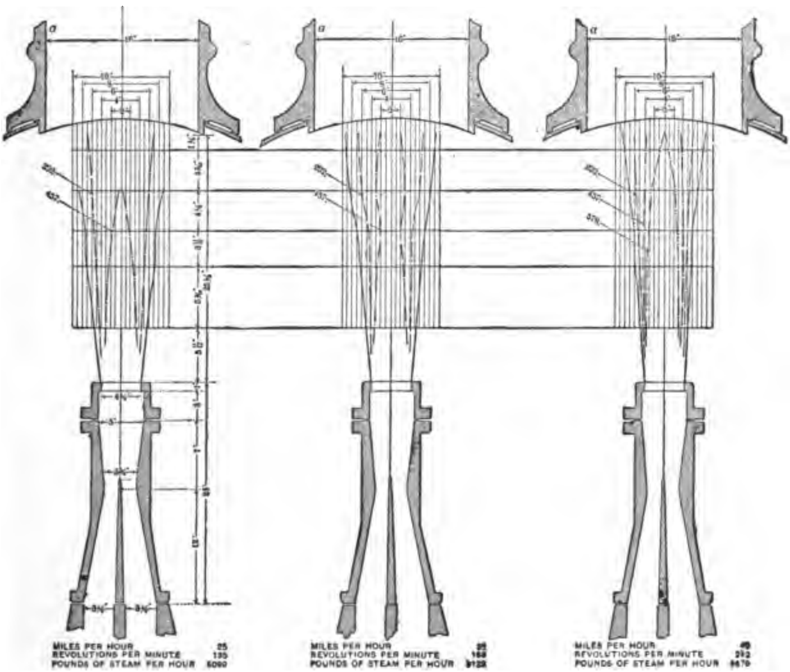


FIG. 110. FIG. 111. FIG. 112.

clusions; first, that the area of the choke should not be less than the area of the nozzle; and, second, that whenever, with a given pipe, it is necessary to sharpen the exhaust action, the result should be secured by contracting the nozzle and not by contracting the choke.

102. Jets Formed by a Steady Blast of Steam.—Figs. 113 to 115 represent jets which were obtained by blocking the slide-valves clear of their seats, and by slightly opening the throttle. That the results thus obtained might be compared with those presented by the earlier figures, the throttle was so adjusted as to pass similar quantities of

steam. Thus, with the locomotive running at 25 miles per hour (Fig. 107) 6090 pounds of steam were exhausted each hour. The steady blow corresponding is represented by Fig. 113, for which the discharge equaled 6400 pounds per hour. The agreement is sufficiently close for practical purposes.

The similarity in the form of the jets serves to fully explain the success of the "blow" in producing draft. A critical comparison of these jets fails to disclose any important difference, except in the

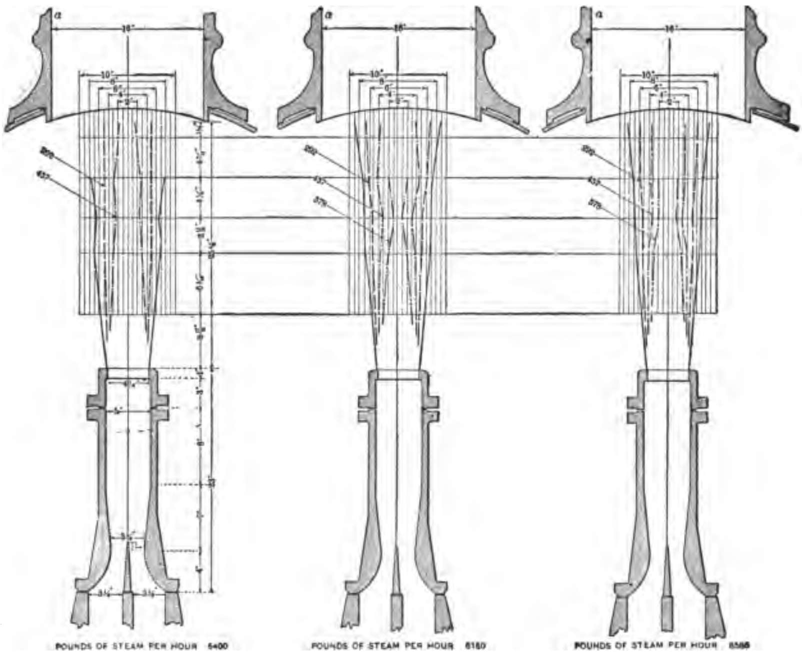
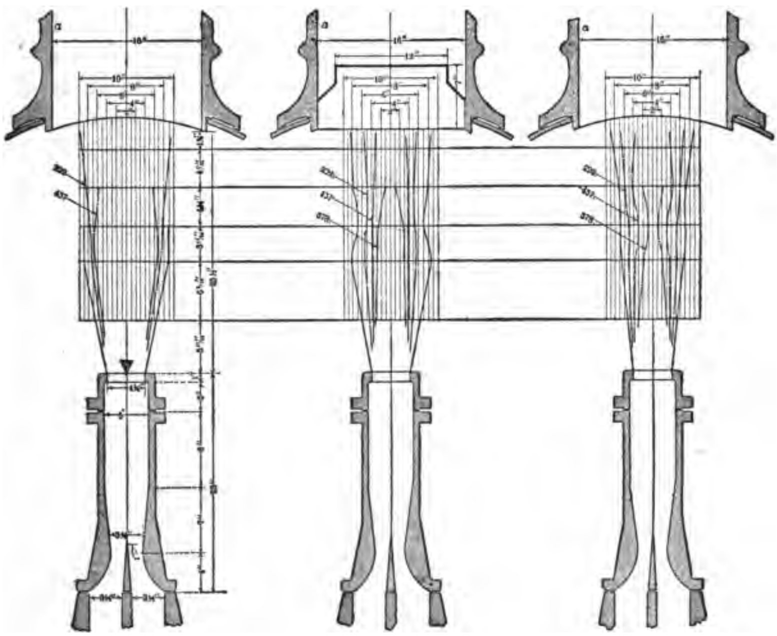


FIG. 113. FIG. 114. FIG. 115.

character of the curves bounding their outline and in that of the velocity curves. The lines in the jets produced by blow are more sinuous than in the jets which are sustained by exhaust section. Corresponding measurements of the draft show that the steady jet possesses substantially the same draft-producing power as the intermittent exhaust, the essential factor being the weight of steam exhausted in a given time.

103. The Form of the Jet as Influenced by Different Tips.—At the time when the Master Mechanics' Committee made its investi-

gation three forms of tips were in common use. These are designated as X, Y, and Z, Fig. 119. The tip X ends in a plain cylindrical portion 2″ in length; the tip Y is contracted in the form of a frustrum of a cone; and the tip Z is in the form of a plain cylinder, ending in an abrupt cylindrical contraction. The forms of the jets delivered by these tips were in each case defined by methods already described. It was found that while the form was nearly the same in all cases, the jet delivered by the tip X has the least divergence, and that of the tip



CROSS BRIDGE — FALSE CHOKE IN THE STACK — STEAM BLOWING THROUGH ENGINE AT REST

FIG. 116. FIG. 117. FIG. 118.

Z the greatest. As to draft-producing qualities, the committee was unable to present convincing proof in favor of any one of the forms experimented upon, the results obtained being practically identical. The opinion is expressed, however, that the highest efficiency will be obtained from the jet which is most dense, and for this reason they were inclined to favor the form X.

104. The Form and Efficiency of the Jet as Affected by Bars over the Tip.—It is well known that engines which refuse to steam may sometimes be made to do so by bridging the exhaust-nozzle with

a small piece of round iron or by a bar or bars having a knife-edged cross-section designed to spread the jet and at the same time impede its motion as little as possible. Experiments were therefore made by the committee of 1896, both upon round bars and upon crosses, shown by Fig. 120, with results confirming the experience on the road. The effect of such a bar upon the form of the jet is shown by Fig. 116, it crowds the interior portions outward. The jet is not made materially larger, but the curves representing different velocities are crowded close together on either side. As to the effect of such a device on the efficiency of the jet, it was found that the draft was improved, but in all cases the presence of the bridge so much increased the back pressure that the efficiency of the front end was reduced. In other words, the presence of the bar or cross produced the same effect upon the back pressure as the substitution of a smaller nozzle without the bars. It was found that cross-bars (Fig. 120), not wider than $\frac{3}{8}$ of an inch, and having the lower portion shaped to a sharp knife-edge, placed on the nozzle, or any distance not greater than one inch above it, increased the back pressure, and that wider bars increased the back pressure at a greater distance from the nozzle. In view of these facts, and also in view of the fact that all evidence goes to show that for highest efficiency the jet should be kept as well compacted as pos-

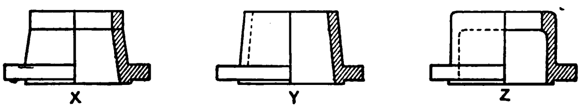


FIG. 119.

sible, the practice of splitting it up was not commended. It is believed to be better practice in cases where the draft is unsatisfactory to reduce the diameter of the exhaust-nozzle than to attempt to secure the desired result by employing a larger nozzle with a bridge above it.

105. The Form of the Jet as Affected by Stack Proportions.— It has already been suggested that the jet adapts itself to the proportions of the stack through which it discharges, and that it shows a disposition to avoid contact with the sides of the stack until very near its top. The facts in the case are better and more definitely defined by Fig. 117, which shows the form of the jet in the presence of a false lining or choke in the stack, so proportioned as to reduce the effective diameter of the stack from 16″, its normal size, to 12″. It is evident that the presence of such a device acts as a throttle on the delivery of the combined stream of gases, and that by so doing it produces a ma-

terial reduction in the velocity of the current within the smoke-box, or points immediately about the jet and below the stack. The reduced velocity allows the stream to broaden out, but it narrows again, and doubtless in some portions of its cross-section its velocity increases as it approaches the base of the contracted stack. Fig. 118 may be employed as a convenient reference in connection with the two preceding figures.

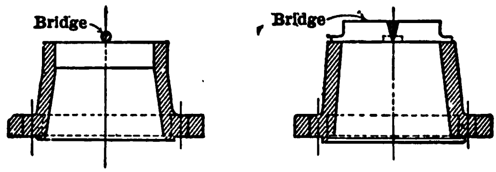


FIG. 120.

106. The Jet as Affected by Cut-off.—For the purpose of determining the effect, if any, upon the form of the jet of changes in cut-off, the load at the locomotive draw-bar was held constant, the speed controlled by throttle-opening, and the engine run with the reverse-lever in several different positions, the form of the jet being determined meanwhile by methods already described. The conclusion drawn from these tests is that changes in the cut-off have no effect upon the form and character of jet, except in so far as they affect the quantity of steam discharged.

107. The Stack Problem.—Prior to 1902 experiments under service conditions involving front-end appliances were but little concerned with the problems of the stack.* This was due to no lack of appreciation of the importance of the stack, but rather to the complicated nature of the problem, and to the fact that the need of information concerning other details had been more urgent. Previous to this time it frequently happened that one locomotive, having twice the cylinder power and boiler capacity of another, would be fitted with a stack of the same size. In this case the furnace gases and exhaust steam of the larger locomotive were re-

* A few years ago there were undertaken at Hanover, Germany, an elaborate series of tests designed to solve the stack problem. The results, which in this country have been known as those of the von Borries-Troske tests, have been widely published. They were conducted upon a full-sized model front end, steam being supplied from a stationary boiler, and air in imitation of the smoke-box gas being admitted to the front end through a restricted opening. While they have undoubtedly served well to guide the practice of German railways, they have not greatly influenced that of our own country.

quired to find their way through an opening which appeared to be not too large for the smaller locomotive. It was, in fact, the practice of some of the important roads to make all stacks of the same diameter, regardless of the size of the locomotive upon which they were to be used. It is evident that in such cases the highest efficiency could only result from a chance combination, and the action in many cases was necessarily far from satisfactory. Moreover, these conditions did not prevail from choice, but rather because the information necessary to guide practice along more logical lines was not available. In view of these facts, the *American Engineer* of New York City, as represented by Mr. G. M. Basford, its editor, having secured the coöperation of a committee of representative motive-power officials,* agreed to become a patron of the locomotive laboratory of Purdue University, for the purpose of securing information which would aid in the logical design of such apparatus.†

108. The Plan of the Tests, as outlined by the laboratory, involved the recognition of the following variable factors:

(*a*) *Form of Stack.*—With reference to this factor, two contours only were provided, one straight and one tapered.

(*b*) *Diameter of Stack.*—Each form of stack was developed into a

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* This committee consists of Mr. Robert Quayle, Chicago & Northwestern; Mr. A. W. Gibbs, Pennsylvania; Mr. G. R. Henderson, Atchison, Topeka & Santa Fé; Mr. F. H. Clark, representing Mr. F. A. Delano, Chicago, Burlington & Quincy; Mr. W. H. Marshall and Mr. H. F. Ball, Lake Shore & Michigan Southern; Mr. F. M. Whyte, representing Mr. A. M. Waitt, New York Central & Hudson River; Mr. W. S. Morris, Chesapeake & Ohio; Mr. C. H. Quereau, Denver & Rio Grande; Mr. C. A. Seley, Norfolk & Western; Mr. J. E. Sague, American Locomotive Company; Mr. E. M. Herr, who was Assistant Superintendent of Motive Power of the Chicago & Northwestern when the Master Mechanics' Association tests of 1896 were made; and Mr. H. H. Vaughan, who, while not then connected with a railroad, was a recognized authority in matters of locomotive design.

† By the arrangement which was entered into, it was agreed that the University should contribute to the proposed work, the equipment of its locomotive-testing laboratory and such portion of the time of its expert staff as could readily be made available; also that the "American Engineer" would meet the cost of special apparatus, of supplies in excess of those usually required in the routine work of the laboratory, and of such additional help as might be necessary. The committee already referred to assisted in an advisory capacity.

A full description of the work thus organized and of the results drawn therefrom will be found running through the issues of the *American Engineer* for the year 1902. The work was carried on in connection with the University's locomotive, "Schenectady No. 2." This is an eight-wheeled engine, weighing 100,000 pounds, having a boiler the least diameter of which is 52 inches.

series of different diameters, the diameters being 9¾ inches, 11¾ inches, 13¾ inches, and 15¾ inches respectively. The dimensions given apply to all portions of the straight stacks above the base, and to the least diameter of the tapered stacks. These diameters were chosen because of their close agreement with the diameters used in the von Borries-Troske experiments.

(c) *Height of Stack.*—Each of the eight stacks already described was made in five sections, the upper four sections each being 10 inches in height. This provision made it possible to employ either of the eight stacks in heights varying from 16½ inches to 56½ inches.

(d) *Exhaust-nozzles.*—It was not expected that the work should involve any investigation of exhaust-pipes or nozzles, this phase of the draft-appliance problem having been covered as already described. It was assumed, however, that in order to get the maximum efficiency of each different diameter and height of stack, it would be necessary to provide a variable height of pipe.

(e) *Power of the Locomotive.*—In the early discussion of the matter the question arose as to whether a combination of apparatus, which would give the highest efficiency when the engine was worked at light power, would prove the most efficient under conditions of heavy power, and that there might be no question arising from this source it was determined to make tests under each combination of apparatus, with the engine running at three different rates of power. It was determined, also, that this could best be accomplished by running all tests under a constant steam pressure, a fixed cut-off, and a wide-open throttle, the desired variations of power being obtained by varying the speed. The conditions chosen were approximately as follows: Steam pressure, 180 lbs.; cut-off, one-third stroke; speeds for different power, 20, 30, and 40 miles respectively; the power under these conditions being approximately 260, 370, and 475. The amount of steam generated by the boiler, and discharged from the exhaust-tip, varies nearly in direct proportion to variations in power.

109. Conditions at the Grate.—It is well known that the draft, as measured by a reduction of pressure within the front end, is affected by conditions other than those which control the action of the steam-jet. With the force of the jet remaining constant, the draft will vary with every change in furnace condition which serves in any way to affect the freedom with which the air is permitted to move through the grate and fire-box. Thus, all other things remaining constant, the draft is reduced by opening the fire-door, and increased by closing the

ash-pan dampers. Similarly, a thin, clean fire, which offers but little resistance to the passage of air, results in light draft, while a thick, heavy fire, which impedes the movement of air at the grate, increases the draft. This being true, it was deemed essential to provide for constant conditions at the grate. As this could not be done in connection with solid fuels, the experimental engine was equipped for burning oil, and the air-openings into the fire-box so arranged that they would at all times be of fixed dimensions. The degree of ease, therefore, with which air found its way into the furnace was

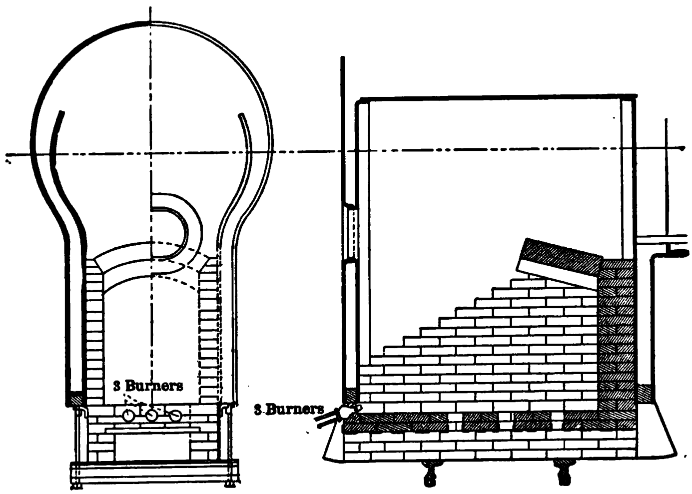


FIG. 121.

unaffected by the condition of fire. Sections of the fire-box as arranged for the tests are shown by Fig. 121.

110. The Experimental Stacks and Nozzles.—The stacks employed in the experiments are shown by Fig. 122. Each stack was made in sections, by the successive removal of which any desired length within the limits shown could be obtained. The several sections were turned to fit one another and were held together by bolts acting upon outside lugs. The nozzles used in connection with the different stacks are shown by Fig. 123, and the details of their construction by Fig. 124. When in use the nozzles were fixed upon a short exhaust-pipe, designed in accord with the recommendation of the Master Mechanics' Committee. They consisted of a wrought-iron body, mounted on a cast-iron base, and fitted at the top with a suit-

able tip. All had the same diameter of tip, namely, 4¼ inches. Notation, in connection with stacks and nozzles, was employed as set

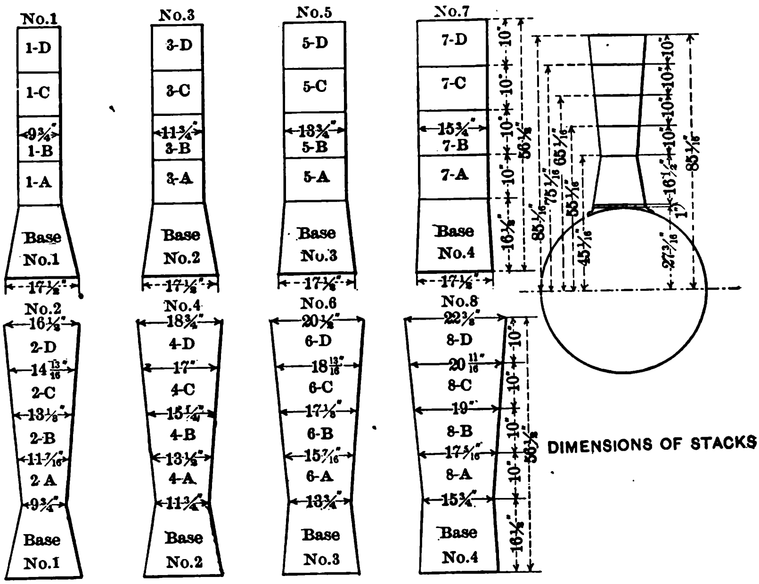


FIG. 122.

forth by Figs. 122 and 123. A No. 7 stack is a straight stack 15¾ inches in diameter. A No. 7 A stack is a stack 26½ inches long,

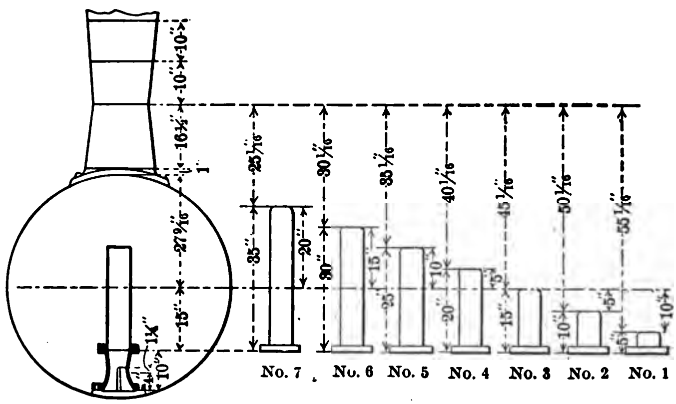


FIG. 123.

while a No. 7 D stack is a stack 56½ inches long. Similarly, a No. 8 A is a taper stack 15¾ inches in diameter at the choke, and 26½ inches

long. The exhaust-pipes and nozzles were numbered. Thus, No. 1 pipe or nozzle is 5 inches high upon a 10-inch base, while nozzle No. 7 is 35 inches high upon a 10-inch base. While these will hereafter be

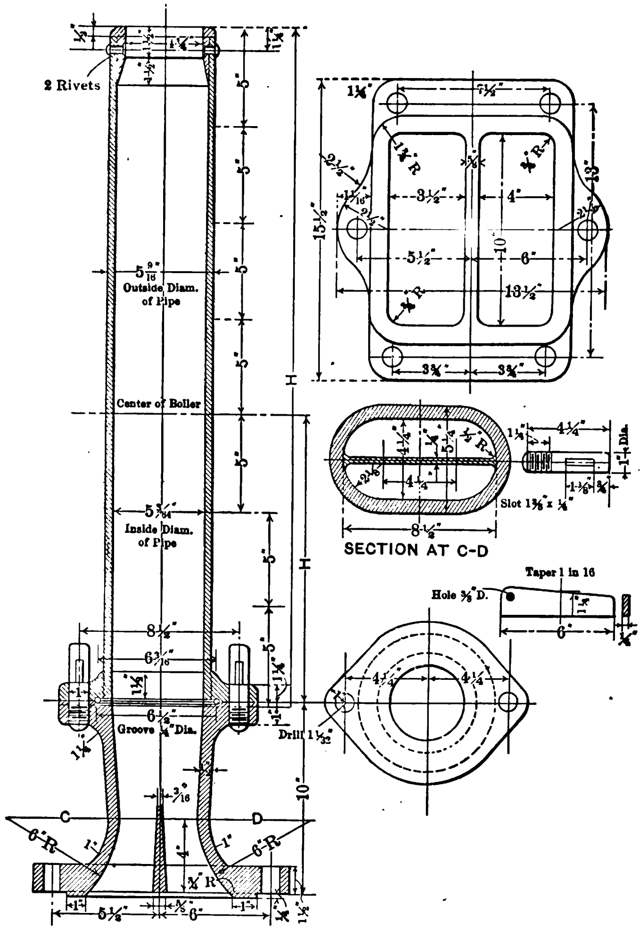


FIG. 124.

referred to as nozzles, they really combine in one design a pipe and nozzle, as will be seen by the drawing representing their construction, Fig. 124. All nozzles were used upon the same base which, as shown, was 10 inches in height.

111. The Tests.—The conditions under which tests were made may be summarized as follows:

VARIABLE SPEED SERIES.

Constants.	Variables.
Boiler pressure, 180. Throttle, wide open. Cut-off, per cent of stroke, 23.8	Speeds: 20, 30, 40, 50 and 60 miles per hour.

VARIABLE CUT-OFF SERIES.

Constants.	Variables.
Boiler pressure, 180. Throttle, wide open. Speed, 40 miles per hour. 194 revolutions per min.	Cut-off, per cent of stroke, 19, 23.8, 26.9, and 35.

With the exception of certain conditions, which could be omitted without interfering with the value of a series, tests were run under each condition specified above for each of 40 different stacks and these again were repeated for from two to seven different heights of exhaust-pipes. To these conditions must also be added others applying to an inside or sliding-stack, the results of which will be separately considered. The work as carried out involved 1032 distinct tests, for each of which the locomotive was brought under specific conditions of running, and for each of which a considerable series of observations were made of permanent record. Besides these recorded tests there have been many other trial or rejected tests.

In anticipation of a series of tests, the engine was warmed by preliminary running. It was then started for a test, and after the throttle and reverse-lever had been brought to their desired positions, the speed of the engine was made to vary by changing its load. When the speed and steam pressure were both that which were required, the signal was given and observations were taken. The engine was then stopped, and such changes in stack or nozzle were made as were necessary for the next set of observations, after which the process above described was repeated. In this manner a day's work consisted in a succession of short runs, with intervals between to allow the change of equipment.

112. Results.—A detailed statement of numerical results, obtained at a speed of 40 miles per hour under a cut-off of 27 per cent, is given in Table LIV While similar tables were developed under the same cut-off for speeds of 20, 30, 50, and 60 miles per hour and under a constant speed of 40 miles per hour, under several different cut-offs, it appears

unnecessary to transfer to these pages the whole of so voluminous a record.*

113. **A Basis of Comparison.**—A careful study of all results made clear the fact that the relative value of the various devices tested could properly be determined by comparing the draft values resulting from their use. That is, that the best stack was that which, under given conditions, gave the best draft. Similarly, that the best height of nozzle was assumed to be that which gave the best draft. This statement ignores the whole question of back pressure, but in the work described this is justified, because of the fact that the changes made as the experiments progressed were found to produce no measurable effect upon the back pressure. While, therefore, in Table LIV. values are given for both back pressure and efficiency, the comparisons which follow are concerned only with the draft.†

* The full record will be found in pages of the *American Engineer and Railroad Journal*, to which references have previously been made.

† In outlining the tests it was proposed to base all comparisons upon the efficiency of the jet, and efficiency was defined as the ratio of back pressure to draft. The assumption of such a measure is based upon the fact that the result which is sought by the use of any combination of draft appliances is a reduction of pressure within the front end, and that the force effecting such a reduction of pressure is a function of the pressure of steam in the passage between the cylinders and the exhaust-tip. The maintenance of considerable pressures in the exhaust-passages tends to reduce the economic performance of the engine, hence it is desired that the necessity for such pressures be, so far as practicable, avoided. The proposed measure of efficiency takes all this into account, for by its use that arrangement of apparatus which will give a desired reduction of pressure in the front end in return for the least back pressure will be the most efficient. Such a conception is perfectly logical. It is not new, but on the contrary is one which has been many times employed in the study of draft appliances.

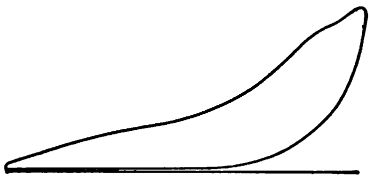
The preceding statement is general in its application. It applies not only to the tests under consideration but to all tests which may be made for the purpose of determining the value of this or that draft appliance. It happens, however, that in the experiments under consideration, the exhaust-tip was of the same size for all tests. Furthermore, it appears as one of the significant results obtained from the tests, that a change in the height of the exhaust-pipe does not affect the back pressure by a measurable amount. Consequently, so far as the present study is concerned, the back pressure for any given condition of running appears as a constant; and the efficiency which, in the general case, is a function of both back pressure and draft, is left to depend upon draft alone. All this being true, it appears that effects resulting from changes in the front-end mechanism, such as were involved by the experiments under consideration, are quite as well shown by a direct comparison of draft values as by a comparison of efficiency values. Moreover, the draft values involve a single observation made under condi-

TABLE LIV.

FORTY-MILE SERIES.

CONSTANTS.

Speed	Miles per hour	40
	R. P. M.	194.4
Pounds of Steam used	Per hour	12988
	Per minute	216
Cut-off	in inches	6.4
	in per cent of stroke	26.9



M.E.P. 52.0 lbs

RESULTS.

Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure. Inches of Water Observed.	Smoke-box Pressure. Pounds Calculated.	Efficiency.	Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure. Inches of Water Observed.	Smoke-box Pressure. Pounds Calculated.	Efficiency.
I.	II.	III.	IV.	V.	VI.	I.	II.	III.	IV.	V.	VI.
	1	2.7	1.2	.043	.016		1	2.65	1.4	.05	.018
	2	2.24	1.2	.0432	.019		2	2.23	1.4	.0504	.023
	3						3	2.52	1.4	.0503	.02
Base No. 1	4	2.25	1.3	.0468	.021	1–C	4	2.05	1.7	.0612	.03
	5	2.25	1.2	.0432	.019		5	2.13	1.8	.0648	.03
	6	2.4	1.2	.0432	.018		6	2.2	1.9	.0684	.031
	7	2.55	1.0	.036	.014		7	2.4	2.0	.072	.03
	1	2.3	1.2	.043	.019		1	2.45	1.3	.047	.019
	2	2.25	1.4	.0504	.022		2	2.25	1.4	.0504	.022
	3	2.50	1.6	.0577	.023		3	2.4	1.4	.0503	.021
1–A	4	2.3	1.7	.061	.027	1–D	4	2.3	1.8	.0648	.028
	5	2.22	1.7	.0612	.028		5	2.22	1.9	.0684	.031
	6	2.2	1.7	.0612	.028		6	2.2	2.0	.072	.033
	7	2.45	1.5	.054	.022		7	2.3	2.1	.0756	.033
	1	2.2	1.4	.05	.023		1	2.6	2.3	.083	.032
	2	2.3	1.4	.0504	.022		2	2.27	2.4	.0864	.038
	3	2.35	1.4	.0503	.021		3	2.35	2.6	.0936	.04
1–B	4	2.25	1.8	.065	.029	2–A	4	2.0	2.4	.0864	.043
	5	2.21	1.9	.0684	.031		5	2.21	2.2	.0792	.036
	6	2.4	1.9	.0684	.028		6	2.4	2.0	.072	.03
	7	2.25	1.9	.0684	.03		7	2.45	1.6	.0578	.024

TABLE LIV.—(*Continued*).

FORTY-MILE SERIES.

Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure. Inches of Water Observed.	Smoke-box Pressure. Pounds Calculated.	Efficiency.	Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure. Inches of Water Observed.	Smoke-box Pressure. Pounds Calculated.	Efficiency
I.	II.	III.	IV.	V.	VI.	I.	II.	III.	IV.	V.	VI.
2-B	1	2.4	3.1	.111	.046	3-C	1	2.5	2.7	.097	.038
	2	2.2	3.2	.1152	.052		2	2.27	2.8	.1008	.044
	3	2.37	3.3	.1188	.05		3	2.34	2.7	.097	.045
	4	2.2	3.0	.108	.049		4	2.3	3.1	.112	.049
	5	2.25	2.9	.1044	.046		5	2.23	3.1	.112	.05
	6	2.4	2.5	.09	.037		6	2.3	3.0	.108	.047
	7	2.4	2.2	.0792	.033		7	2.4	2.8	.101	.042
2-C	1	2.45	3.5	.126	.051	3-D	1	2.6	2.4	.086	.033
	2	2.15	3.8	.1368	.064		2	2.23	2.7	.0972	.044
	3	2.38	3.7	.1332	.056		3	2.6	3.0	.108	.041
	4	2.3	3.5	.126	.055		4	2.4	3.0	.108	.045
	5	2.25	3.5	.126	.056		5	2.2	3.0	.108	.049
	6	2.2	3.1	.1116	.051		6	2.2	3.1	.112	.051
	7	2.2	2.7	.0972	.044		7	2.4	3.2	.115	.048
2-D	1	2.45	4.0	.144	.059	4-A	1	2.3	3.0	.108	.047
	2	2.1	4.3	.1548	.074		2	2.3	3.3	.1188	.052
	3	2.42	4.0	.144	.059		3	2.35	3.0	.108	.046
	4	2.4	3.9	.1404	.058		4	2.2	2.9	.104	.047
	5	2.21	4.0	.144	.065		5	2.3	2.5	.09	.039
	6	2.4	3.4	.122	.051		6	2.2	2.0	.072	.033
	7	2.25	3.2	.1152	.051		7	2.4	1.4	.05	.021
Base No. 2	1					4-B	1	2.2	3.5	.126	.057
	2	2.33	2.1	.0756	.032		2	2.35	4.1	.1476	.063
	3						3	2.34	3.4	.122	.052
	4	2.3	1.9	.068	.030		4	2.25	3.6	.130	.058
	5	2.22	1.6	.058	.026		5	2.27	3.2	.152	.051
	6	2.2	1.2	.043	.019		6	2.4	2.7	.097	.041
	7	2.5	.9	.032	.013		7	2.6	2.1	.076	.029
3-A	1	2.6	2.3	.082	.032	4-C	1	2.5	4.3	.155	.062
	2	2.4	2.6	.094	.039		2	2.3	4.6	.1656	.072
	3	2.4	2.65	.096	.039		3	2.28	4.0	.144	.063
	4	2.2	2.6	.094	.043		4	2.3	4.0	.144	.063
	5	2.23	2.4	.086	.038		5	2.34	3.7	.133	.057
	6	2.2	2.1	.076	.034		6	2.4	3.0	.108	.045
	7	2.5	1.7	.061	.025		7	2.3	2.6	.094	.041
3-B	1	2.5	2.7	.097	.039	4-D	1	2.8	5.3	.191	.068
	2	2.3	2.6	.094	.041		2	2.27	5.0	.18	.079
	3	2.5	2.9	.104	.042		3	2.41	4.4	.158	.065
	4	2.3	2.9	.104	.045		4	2.4	4.3	.155	.065
	5	2.25	2.9	.104	.046		5	2.4	4.2	.151	.063
	6	2.2	2.6	.094	.043		6	2.4	3.8	.136	.057
	7	2.65	2.4	.087	.033		7	2.35	3.2	.115	.049

TABLE LIV.—(*Continued*).

FORTY-MILE SERIES.

Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure.		Efficiency.	Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure.		Efficiency.
			Inches of Water Observed.	Pounds Calculated.					Inches of Water Observed.	Pounds Calculated.	
I.	II.	III.	IV.	V.	VI.	I.	II.	III.	IV.	V.	VI.
	1						1	2.27	4.0	.144	.064
	2	2.35	2.8	.1008	.043		2	2.33	4.2	.151	.065
	3						3	2.2	4.2	.151	.068
Base No. 3	4	2.45	1.9	.0684	.028	6–B	4	2.3	3.4	.1224	.053
	5						5	2.24	3.0	.108	.048
	6	2.45	1.1	.0396	.016		6	2.15	2.4	.086	.040
	7	2.25	.7	.0252	.011		7	2.25	1.9	.0684	.030
	1	2.41	3.2	.115	.048		1	2.1	4.4	.158	.075
	2	2.33	3.5	.126	.054		2	2.26	4.6	.1656	.073
	3	2.4	3.1	.1116	.046		3	2.2	4.5	.162	.074
5–A	4	2.4	2.8	.1008	.042	6–C	4	2.35	3.9	.1404	.06
	5	2.25	2.4	.0864	.038		5	2.27	3.4	.1224	.056
	6	2.2	2.0	.072	.033		6	2.2	2.9	.104	.047
	7	2.35	1.4	.0503	.021		7	2.25	2.4	.0684	.038
	1	2.37	3.2	.115	.049		1	2.1	4.8	.173	.082
	2	2.4	3.9	.1402	.058		2	2.25	5.0	.18	.08
	3	2.36	3.4	.1221	.052		3	2.3	5.2	.1872	.081
5–B	4	2.45	3.2	.1152	.047	6–D	4	2.4	4.4	.1584	.066
	5	2.25	3.2	.1152	.051		5	2.2	4.1	.1476	.067
	6	2.3	2.8	.1008	.044		6	2.25	3.3	.1188	.053
	7	2.4	2.8	.0792	.033		7	2.5	2.8	.1008	.04
	1	2.3	3.2	.115	.05		1				
	2	2.25	3.6	.1296	.058		2	2.30	2.8	.1	.043
	3	2.4	3.6	.130	.054		3				
5–C	4	2.4	3.7	.1332	.055	Base No. 4	4	2.3	1.7	.061	.027
	5	2.25	3.6	.1296	.057		5				
	6	2.25	3.4	.122	.054		6	2.3	.75	.027	.012
	7	2.4	3.1	.1116	.046		7	2.35	.5	.018	.008
	1	2.45	3.4	.122	.05		1	2.22	3.4	.122	.055
	2	2.26	3.8	.1368	.059		2	2.35	3.6	.129	.055
	3	2.3	3.6	.13	.056		3	2.13	3.0	.108	.051
5–D	4	2.45	4.0	.144	.054	7–A	4	2.33	2.7	.097	.042
	5	2.22	3.9	.1404	.063		5	2.25	2.4	.086	.038
	6	2.3	3.8	.1368	.059		6	2.3	1.6	.057	.025
	7	2.35	3.5	.126	.054		7	2.3	1.1	.039	.017
	1	2.4	3.6	.129	.054		1	2.18	3.6	.129	.059
	2	2.3	3.5	.126	.055		2	2.25	4.2	.158	.07
	3	2.2	3.4	.122	.055		3	2.30	3.6	.130	.056
6–A	4	2.45	2.7	.0972	.04	7–B	4	2.4	3.4	.122	.051
	5	2.28	2.4	.0864	.036		5	2.22	3.0	.108	.049
	6	2.4	1.8	.0648	.027		6	2.2	2.4	.086	.039
	7	2.4	1.3	.0468	.019		7	2.2	1.9	.068	.031

TABLE LIV.—(*Continued*).

FORTY-MILE SERIES.

Stack.	Nozzle.	Observed Back Pressure, Lbs.	Smoke-box Pressure. Inches of Water Observed.	Smoke-box Pressure. Pounds Calculated.	Efficiency.
I.	II.	III.	IV.	V.	VI.
7–C	1	2.21	3.6	.129	.058
	2	2.35	4.6	.166	.071
	3	2.37	4.0	.144	.061
	4	2.35	3.8	.136	.058
	5	2.22	3.8	.136	.062
	6	2.3	3.2	.1156	.05
	7	2.45	2.9	.104	.043
7–D	1	2.21	3.9	.14	.067
	2	2.4	4.7	.169	.070
	3	2.3	4.2	.151	.066
	4	2.2	4.2	.151	.069
	5	2.25	4.2	.151	.067
	6	2.35	3.7	.133	.057
	7	2.35	3.5	.126	.054
8–A	1	2.22	3.5	.126	.057
	2	2.3	3.6	.129	.056
	3	2.4	3.0	.108	.045
	4	2.35	2.3	.082	.035
	5	2.33	1.9	.068	.029
	6	2.30	1.25	.045	.02
	7	2.35	.9	.032	.014
8–B	1	2.25	4.1	.148	.066
	2	2.4	4.2	.151	.063
	3	2 45	3.5	.126	.051
	4	2.35	2.9	.104	.045
	5	2.5	2.4	.086	.035
	6	2.4	1.9	.068	.028
	7	2.25	1.4	.05	.022
8–C	1	1.27	4.5	.162	.071
	2	2.25	4.4	.158	.07
	3	2.4	4.0	.144	.06
	4	2.25	3.4	.122	.054
	5	2.7	3.0	.108	.037
	6	2.4	2.4	.086	.036
	7	2.2	1.9	.068	.031
8–D	1	2.4	4.9	.176	.073
	2	2.23	4.7	.169	.075
	3	2.4	4.2	.151	.069
	4	2.4	3.6	.130	.054
	5	2.8	3.6	.129	.036
	6	2.4	2.7	.097	.04
	7	2.25	2.3	.082	.036
Normal, Petticoat Pipe In	*	2.4	3.9	.14	.058
Normal, Petticoat Pipe Out	*	2.4	3.8	.136	.057
Sliding A	1	2.8	3.6	.129	.046
	2	2.2	3.0	.108	.049
	3	2.3	2.7	.097	.042
Sliding B	1	2.6	3.0	.108	.041
	2	2.2	3.4	.122	.055
	3	2.3	3.1	.111	.048
Sliding C	1	2.6	3.5	.126	.048
	2	2.3	3.9	.140	.061
	3	2.5	4.2	.151	.06
Sliding D	1	2.4	4.0	.144	.06
	2	2.2	4.1	.147	.067
	3	2.4	4.0	.144	.06

* Normal.

114. The Effect upon Stack Proportions of Changes in Speed and Cut-off.—The experiments were made to involve a great variety of conditions with reference to speed and cut-off, in order that the differences in the efficiency of the stack arrangement, which resulted from such changes, might be known. From a careful study of all of this data it can be stated with confidence that, within reasonable limits, no change in the condition of running affects in any marked degree the efficiency of the stack; that a stack and nozzle which are satisfactory at

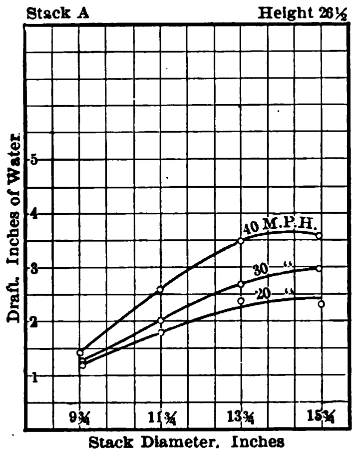


FIG. 125.—Straight Stack, Nozzle No. 2.

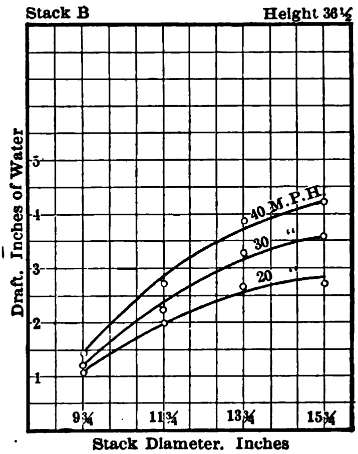


FIG. 126.—Straight Stack, Nozzle No. 2.

one speed will be found satisfactory at all speeds, and if satisfactory under one condition of cut-off they will be found reasonably satisfactory under all conditions of cut-off. This conclusion, which is of the highest importance, cannot be perfectly demonstrated without the use of an elaborate display of data, but its truth is well illustrated by the similarity of the curves representing the draft obtained under three different speeds by means of several different stacks, Figs. 125 to 132.*

tions favorable to accuracy, and, consequently, they supply a better basis for comparison than efficiency, which depends on two observations, one of which is rather difficult to obtain.

* These figures have been chosen to illustrate the effect upon the draft of changes in the proportions of the stack and height of the nozzle. Figs. 125 to 128 present results obtained by using straight stacks of various heights and diameters, in connection with an exhaust tip 2 inches below the center of the boiler Figs. 129 to 132 show results obtained by the use of tapered stacks and an exhaust tip 20 inches above the center of the boiler. The steam pressure, throttle opening, and cut-off were the same for all tests represented by these diagrams.

Curves representing results obtained at different cut-offs present the same characteristics. With a given weight of steam discharged,

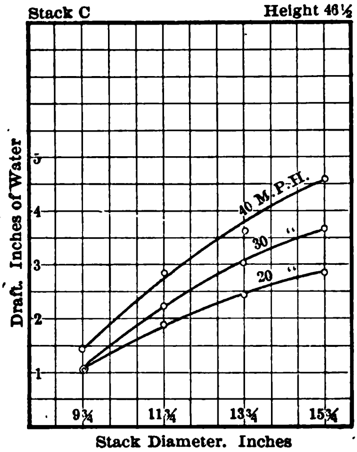


FIG. 127.—Straight Stack, Nozzle No. 2.

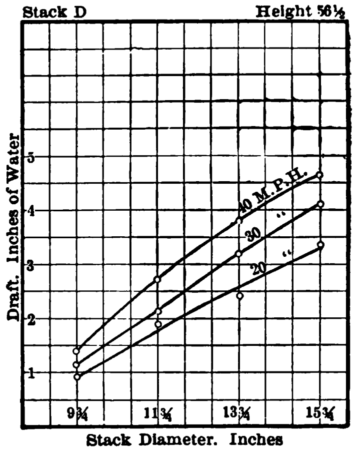


FIG. 128.—Straight Stack, Nozzle No. 2.

whether in the heavy exhausts incident to slow speed, or the more rapid impulses which are sent forth at higher speed, the draft result-

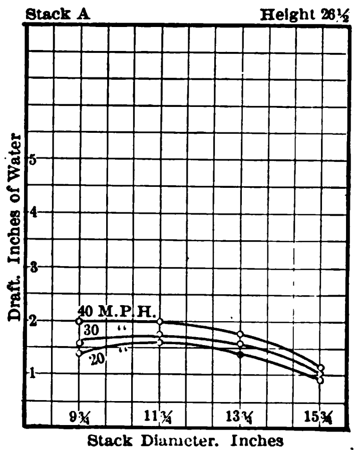


FIG. 129.—Tapered Stack, Nozzle No. 7.

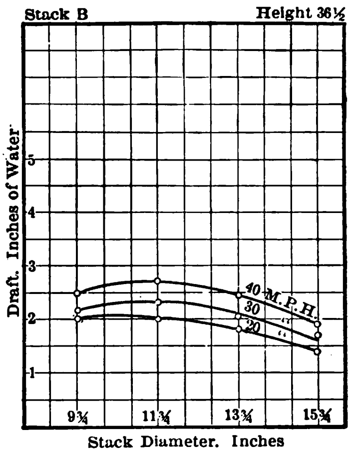


FIG. 130.—Tapered Stack, Nozzle No. 7.

ing is practically the same. But whenever the weight of steam discharged per minute changes, the draft will change. That this is true for changes of speed is made clear by Table LV., in which is given a

comparison of changes in draft values with corresponding changes in the volume of steam exhausted.

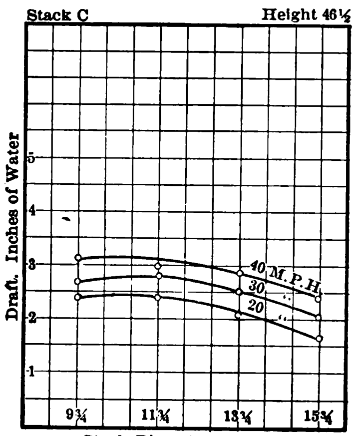


Fig. 131.—Tapered Stack, Nozzle No. 7.

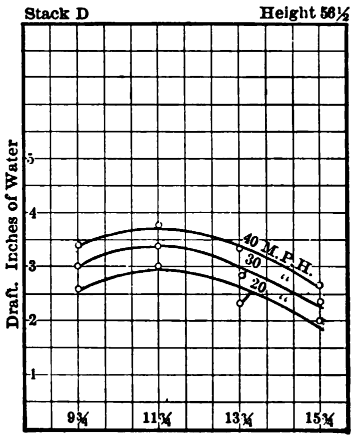


Fig. 132.—Tapered Stack, Nozzle No. 7.

Table LV.

A COMPARISON OF CHANGES IN DRAFT VALUES WITH CORRESPONDING CHANGES IN THE VOLUME OF STEAM EXHAUSTED.

Nozzle No. 3. Exhaust-tip on Center Line of Boiler.

Stacks.			Change in Speed, Miles.	Ratio of Change in Draft.	Ratio of Change in Steam Used per Hour.
Height.	Diameter.	Form.			
26½	9¾	Taper	20 to 30	1.25	1.33
26½	9¾	Taper	30 to 40	1.30	1.19
36½	9¾	Taper	20 to 30	1.30	1.33
36½	9¾	Taper	30 to 40	1.23	1.19
46½	9¾	Taper	20 to 30	1.15	1.33
46½	9¾	Taper	30 to 40	1.19	1.19
56½	9¾	Taper	20 to 30	1.25	1.33
56½	9¾	Taper	30 to 40	1.14	1.19
26½	11¾	Taper	20 to 30	1.23	1.33
26½	11¾	Taper	30 to 40	1.14	1.19
36½	11¾	Taper	20 to 30	1.29	1.33
36½	11¾	Taper	30 to 40	1.13	1.19
46½	11¾	Taper	20 to 30	1.24	1.33
46½	11¾	Taper	30 to 40	1.11	1.19
56½	11¾	Taper	20 to 30	1.26	1.33
56½	11¾	Taper	30 to 40	1.15	1.19
		Average. . .	. . . .	1.21	1.26

Following the first line of this table, it will be seen that when the speed is changed from 20 to 30 miles, the draft is increased in the ratio

of from 1 to 1.25. and the steam exhausted is increased in the ratio of from 1 to 1.33, which is an approach to agreement. Similar comparisons are shown for various diameters and heights of stacks, the average of all being, for changes in draft, 1.21, and for changes in steam exhausted, 1.26, which makes the check very close. The result of comparisons involving other values tends to further confirm the general conclusion.

115. A Review of Best Results.—From an inspection of the results of all the tests, the highest draft readings have been selected for each condition of speed and cut-off. These, with the designation of stacks and nozzle employed in securing them, are set down in the columns of Table LVI. Results thus chosen constitute approximately 5 per cent of the whole number obtained, and may be accepted as representing the best results obtainable under any combination of stack and nozzle involved by the experiments.

TABLE LVI.

STACK AND NOZZLE COMBINATION GIVING BEST RESULTS.

Speed, Miles per Hour.	Cut-off Notch.	Draft, Inches of Water.	Nozzle Number.	Stack Number.	Speed, Miles per Hour.	Cut-off Notch.	Draft, Inches of Water.	Nozzle Number.	Stack Number.
20	5	3.4	2	7-D	50	5	5.5	1	8-D
20	5	3.3	1	6-D	50	5	5.4	1	6-D
20	5	3.3	2	4-D	50	5	5.4	2	6-D
20	5	3.2	3	7-D	50	5	5.4	3	8-D
20	5	3.1	5	4-D	50	5	5.3	1	4-D
20	5	3.1	2	6-D	60	5	6.6	2	4-D
20	5	3.1	3	6-D	60	5	6.4	1	8-D
20	5	3.1	4	6-D	60	5	6.2	1	6-D
20	5	3.1	2	8-D	60	5	6.1	1	4-D
30	5	4.3	2	4-D	60	5	6.0	2	8-D
30	5	4.3	1	6-D	40	3	3.6	2	8-D
30	5	4.1	2	7-D	40	3	3.2	1	8-D
30	5	4.0	2	6-D	40	3	3.1	1	4-D
30	5	4.0	3	7-D	40	3	3.0	3	6-D
30	5	4.0	3	7-D	40	3	3.0	3	8-D
30	5	4.0	1	8-D	40	3	3.0	1	6-C
30	5	4.0	2	8-D	40	5	5.3	1	4-D
30	5	4.0	3	8-D	40	5	5.0	1	8-D
40	5	5.2	3	6-D	40	5	4.8	2	8-D
40	5	5.0	2	4-D	40	5	4.7	1	6-D
40	5	5.0	2	6-D	40	5	4.6	1	8-C
40	5	4.9	1	8-D	40	7	7.8	1	8-D
40	5	4.8	1	6-D	40	7	7.6	1	4-D
40	5	4.7	2	7-D	40	7	7.6	1	6-D
40	5	4.7	2	8-D	40	7	7.6	2	8-D
40	5	4.6	2	4-C	40	7	7.4	2	6-D

This table shows at a glance that all the highest draft readings were obtained with the D stacks, which designation embraces those of

greatest length employed in the experiments (56½ inches). The C stacks, which are 10 inches shorter than the D stacks, appear in the table but three times, and in two of these instances the draft is inferior to that given by the higher stack. Stacks bearing even numbers are tapered, and those bearing odd numbers are straight. It is noteworthy that practically all of the numbers appearing in the table of best results represent tapered stacks. The stack numbers which most frequently appear are 4, 6, and 8, representing a diameter at the choke of 11¾, 13¾, and 15¾ inches respectively.

Resorting now to a detailed study covering all the various heights and diameters of stacks experimented upon, and dealing first with the straight stacks, the large spots upon Figs. 133 to 136, inclusive, show the best diameters for each height of stack experimented upon, and for each height of nozzle employed. Thus, disregarding for the present the oblique line drawn upon these diagrams, the large spots upon Fig. 133 show the best diameter of stack for each height of nozzle employed, when the height of the stack is limited to 26½ inches; those upon Fig. 134 show the best diameter of stack for each height of nozzle employed, when the height of stack is limited to 36½ inches. In each case the large spots represent the diameter of the experimental stacks giving the highest draft. The points are not located from curves. Since the experimental stacks varied one from another by steps of 2 inches, the exact diameter represented by a given large spot does not necessarily represent the most desirable diameter for the conditions defined; that we will hereafter proceed to find.

Continuing to give attention to the black spots of the diagrams, it will be noticed that some of the larger spots have smaller spots connected with them by a horizontal line. In some cases there is a small spot on one side of the larger spot, and in other cases there are smaller spots on both sides. These smaller spots indicate that the next sized stack on one or the other sides, or on both sides of the best experimental stack gave results almost as good as the best. Instead of having a series of points representing the experimental data, we have a series of lines which may be employed to establish a zone of good performance. Concerning the width of this zone, it should be noted that the lines span but half the distance between the position of the stack which was best and that which is almost as good; also, that in cases where stacks on one or the other side of the best were not almost as good, no line whatever has been drawn.

A review of the diagrams, Figs. 133 to 136, shows at a glance that

the largest straight stack ($15\frac{3}{4}$ inches), while, perhaps, sufficient in diameter for the least height (Fig. 133), was quite insufficient for the lower nozzle position when the stack height was increased beyond $26\frac{1}{2}$ inches. This is best shown by Fig. 136. The best results in connection with this stack were obtained for the four lowest positions of the

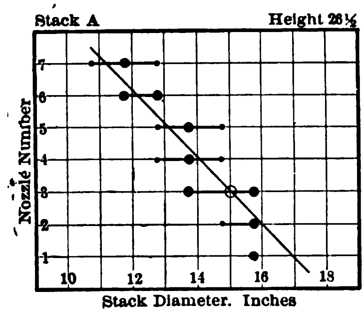


FIG. 133.

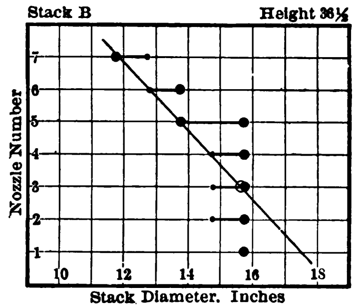


FIG. 134.

exhaust-pipe. When the tip of the exhaust-pipe was 5 inches above the center line of the boiler, the next smaller stack was almost as good, but for the lower nozzles (3, 2, and 1), the largest stack was in a marked degree better than any which were smaller, justifying the conclusion that if a stack as large as 18 inches in diameter had been tried, the

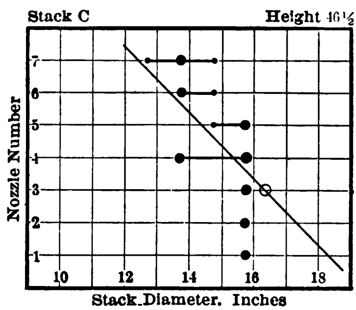


FIG. 135.

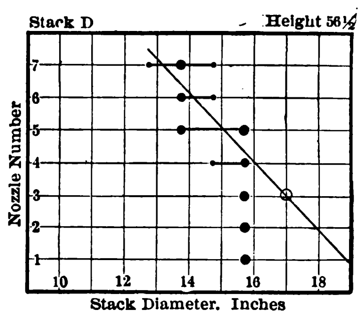


FIG. 136.

results obtained therefrom would have been given a place on the diagram.

Best results obtained from tapered stacks, plotted in a manner already described in considering the action of the straight stack, are presented as Figs. 137 to 140 inclusive. In this case, however, the several diameters of stack experimented upon cover a range sufficiently

wide to permit the selection of the best stack for all heights of stacks in combination with all heights of nozzles.

116. Relation of Height to Diameter of Stack.—The problem of stack design, as disclosed by the data already presented, is not to be regarded as one requiring a high degree of refinement. The data show

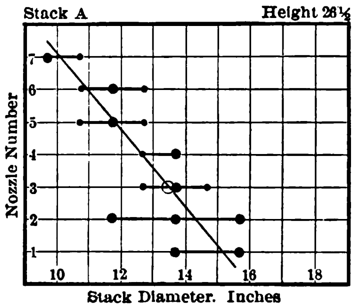


FIG. 137.

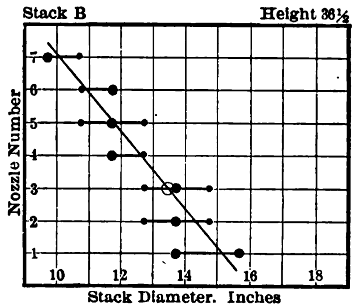


FIG. 138.

that two stacks varying as much as 2 inches in diameter sometimes give results which are almost identical. It happens in some cases, also, that a stack of a diameter which gives maximum results is almost equaled in its performance by a stack 2 inches less, and also by a stack 2 inches greater in diameter. In such a case a variation of 4 inches

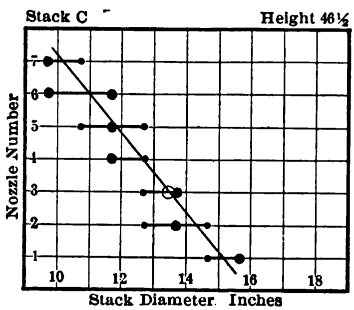


FIG. 139.

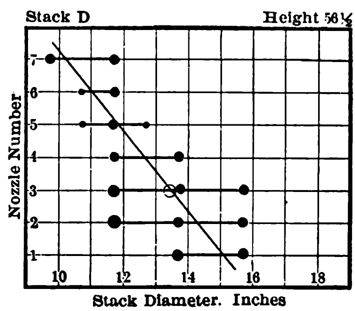


FIG. 140.

in the diameter of the stack appears not to be significant. This is only true, however, with certain heights of stacks in combination with certain heights of nozzles, and, as will be hereafter shown, the occurrence of such cases is more frequent in the case of tapered stacks than in the case of straight stacks.

It is evident, therefore, that it will be impracticable to determine within a small fraction of an inch the diameter of stack corresponding

with any given height of stack and position of nozzle, which can be said to accurately represent the experimental results, for it is sometimes hard to say whether one or another of two stacks is the better. For the purpose, therefore, of reducing the various discordant elements to order, that a general law may be formulated, no harm will be done by employing a fair degree of liberality in interpreting the data.

Straight Stacks.—In order that the best relation of diameter and height of stack may be stated, it will at first be necessary to eliminate variations in the height of the nozzles. Comparisons will, therefore, first be based upon results obtained with nozzle No. 3, this being the nozzle for which the tip is on the center line of the boiler. From a study of the plotted spots of Fig. 133, it has been assumed that, when the exhaust-tip is on the center of the boiler, and the height of the stack is 26½ inches, the most satisfactory diameter, as disclosed by the data, is 15 inches, and a circle has been struck on the diagram (Fig. 133) to represent this value. Similarly, from Fig. 134, it appears that when the exhaust-tip is on the center of the boiler and the height of the stack is 36½ inches, the most satisfactory diameter is 15.66, and a circle, the location of which corresponds with this value, has been struck on this diagram. In the same manner for the 46½-inch stack, the circle has been struck to represent a diameter of 16.33 inches, and for the 56½-inch stack the circle has been struck to represent a diameter of 17 inches. These, then, are assumed to be the best diameters of stacks for each of the several heights experimented upon, when the exhaust-nozzle is on the center line of the boiler.

Stating these facts in the form of equations, in which d is the diameter of stack in inches, we have, for the engine experimented upon, the following:

For straight stacks 26½ inches high:

$$d_1 = 15 = .28 \times 54.$$

For straight stacks 36½ inches high:

$$d_2 = 15.66 = .29 \times 54.$$

For straight stacks 46½ inches high:

$$d_3 = 16.33 = .30 \times 54.$$

For straight stacks 56½ inches high:

$$d_4 = 17 = .31 \times 54.$$

For all of these values the exhaust-tip was on the center of the boiler, and the diameter of the front end of the boiler experimented upon was 54 inches. If, now, we may assume that the data obtained from the engine experimented upon is applicable to engines having boilers of different diameters, and if we may assume also that in applying the data to other engines, we may use the diameter of the boiler as a unit of measure, then the diameter of stack for any boiler whatsoever which has the exhaust-tip on the center line should be represented by equations in which D, the diameter of the inside of the front end, is substituted for the value 54 in the preceding equations. The result is as follows:

For straight stacks 26½ inches high:

$$d_1 = 15 = .28 \times D.$$

For straight stacks 36½ inches high:

$$d_2 = 15.66 = .29 \times D.$$

For straight stacks 46½ inches high:

$$d_3 = 16.33 = .30 \times D.$$

For straight stacks 56½ inches high:

$$d_4 = 17 = .31 \times D.$$

If, now, an expression can be found which maybe substituted for the coefficient of D, and which will represent the height of stack in inches in each of the four equations, it will be possible to write a single equation in the place of four. That this may be more readily accomplished, the values representing best diameters with which we have been dealing were so chosen that, while doing no violence to the experimental data, they will, when plotted in terms of height of stack, fall upon the same straight line, all as shown by Fig. 141. This fact makes it possible to write in simple form a general equation expressing the relation thus defined. Thus, by Fig. 141, it is apparent that when the stack height is zero, the diameter is equal to something over 13 inches (more exactly, 13.23), and the slope of the line connecting the several experimental points is such that with each inch height of stack, the diameter increases .00123 inch$\times$54. We may therefore write as the coefficient of D, in the four preceding equations,

$$(.246 + .00123H),$$

in which H is the height of the stack in inches.

We may, therefore, write for any straight stack when the exhaust-nozzle is on the center line of the boiler,

$$d = (.246 + .00123H)D;$$

d being the diameter of the stack in inches when the exhaust-nozzle is on the center line of boiler, H the height of the stack in inches, and D the diameter of the front end of the boiler in inches. Modification in the form of this equation to satisfy the condition arising from varying heights of exhaust-nozzle will be hereafter considered.

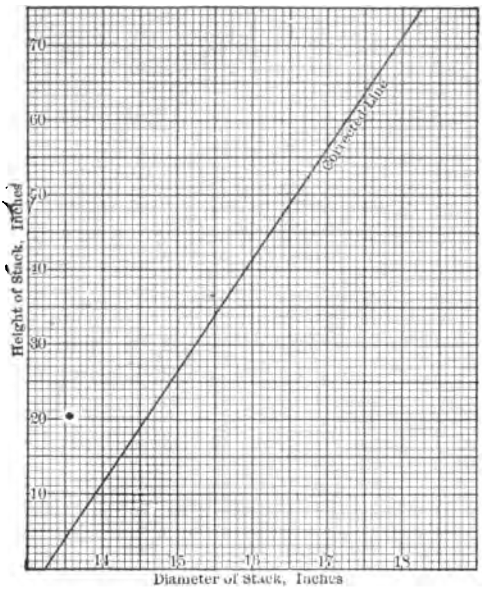


FIG. 141.

Tapered Stacks.—The best results attending the use of the tapered stacks of each different height experimented upon, in connection with the seven different heights of exhaust-nozzles, appear in Figs. 137 to 140 inclusive. In these diagrams the experimental results are shown by means of black spots connected by horizontal lines in the manner already described in connection with the straight stack. When two stacks give equally good results, both points are located and the spots connected by a horizontal line; and where a larger or smaller stack gives results almost as good as the best, a line is extended in its direction, terminating in a small spot midway between that representing

the best stack and the position representing the stack which is almost as good.

Proceeding, as in the case of the straight stacks, to locate a representative point in line with nozzle No. 3, which will fairly represent the experimental data, choice has been made of the diameter 13½ inches, and a circle drawn upon all diagrams while at this diameter. It appears, therefore, that an important conclusion to be derived from the experimental data is to the effect that a tapered stack having a least diameter of 13½ inches gives maximum results for all heights of stack between the limits of 26½ and 56½ inches. In other words, unlike the straight stack, the diameter of the tapered stacks does not need to be varied with changes in the height.

Stating this fact in the form of an equation, therefore, we have for a tapered stack upon the Purdue engine, the diameter of the boiler of which is 54 inches, the following:

$$d_a = 13.5;$$

also $$d_a = .25 \times 54 \text{ inches}.$$

Assuming that the results thus obtained from the experimental engine may be applied to other engines having different diameters of boilers, and using the diameter of the boiler as a unit of measure, we may write for all locomotives, and for all heights of stack where the exhaust-tip is on the center line of the boiler:

$$d = .25D,$$

in which d is the least diameter of the tapered stack when the exhaust-tip is on the center line of the boiler, and D is the diameter of the front end of the boiler.

117. The Effect of Changes in the Height of the Exhaust-nozzle upon the Diameter of the Stack.—For the purpose of passing from the results obtained from the experimental engine to those to be expected from engines having boilers of other diameters, using the boiler diameter as a unit of measure, it has been necessary thus far to deal with conditions for which the parts are symmetrically arranged. It is for this reason that the central position of the nozzle is the only position which has been employed. We may now consider the influence upon the diameter of the stack resulting from changes in the height of the nozzle.

The points represented by the circles (Figs. 133 to 140), and which

have been the basis of equations thus far written, have been so located that it is possible to draw through each of them a straight line, which will fairly represent the best diameter of stack for all heights of nozzles. The oblique line which appears in the several figures may be regarded as such a line. It now remains to find an expression for this line which can be added, as a new term, to the equation which has already been deduced, for the purpose of modifying the final results as demanded by the differences in results obtained when changes are made in the height of the nozzle.

Straight Stacks.—In Figs. 133 to 136 inclusive, representing the straight stacks, the oblique lines representing the relationship between diameter of stack and height of nozzle, as disclosed by the experimental data, have all been drawn at a constant angle. The slope of the line is such that, assuming the effect of the nozzle in position 3 to be zero, the effect upon the diameter of the stack of each inch change in the height of the exhaust-nozzle equals .19 of an inch. It is evident that this correction will effect an increase in the diameter of the stack when the nozzle position is below the center line of the boiler, and a decrease in the diameter of the stack when the exhaust-nozzle is above the center line of the boiler. We may, therefore, write as a new term in the equation, giving the best diameter of a straight stack,

$$.19h,$$

in which h is the distance in inches between the center line of the boiler and the exhaust-tip, the sign preceding this term being positive when h is the distance below the center line, and negative when h is the distance above the center line.

Tapered Stacks.—By a similar process it may be shown that the effect of changes in the height of nozzle on the diameter of a tapered stack is expressed by

$$.16h,$$

the sign preceding the term being positive when h is the distance below the center line, and negative when h is the distance above the center line.

118. Equations giving Stack Diameters for any Height of Stack between the Limits of 26 and 56 Inches, and any Height of Nozzle between the Limits of 10 Inches below the Center of the Boiler, and 20 Inches above the Center of the Boiler, and for any Diameter of Front End.—Combining the expressions of the two preceding paragraphs, we may have equations giving diameter of stack

in terms of its height, diameter of front end, and the distance between the center line of the boiler and the top of the exhaust-tip. These several equations obviously are the equations of the oblique lines appearing in the corresponding diagrams, Figs. 133 to 140. They are as follows:

For straight stacks:

When the exhaust-nozzle is below the center line of the boiler

$$d = (.246 + .00123H)D + .19h.$$

When the exhaust-nozzle is above the center line of the boiler

$$d = (.246 + .00123H)D - .19h.$$

When the exhaust-nozzle is on the center line, h is equal to zero and the equation becomes

$$d = (.246 + .00123H)D.$$

For tapered stacks:

When the nozzle is below the center line of the boiler

$$d = .25D + .16h.$$

When the nozzle is above the center line of the boiler

$$d = .25D - .16h.$$

When the nozzle is on the center line of the boiler, h becomes zero, and

$$d = .25D.$$

In all of these equations d is the diameter of the stack in inches. For the tapered stack it is the least diameter or diameter of choke. H is the height of stack in inches, and for maximum efficiency should always be given as large a value as conditions will admit. D is the diameter of the front end of the boiler in inches, and h the distance between center line of boiler and the top of the exhaust-tip.

If D in the several equations is made equal to 54, the diameter of the front end of the Purdue locomotive, the equations will give results identical with those which are assumed to represent the maximums obtained in the course of the experiments. How far they should be employed in the manner which has been indicated for an engine having a boiler different in size cannot, of course, be stated with certainty,

though it seems clear that, where the conditions surrounding stack and nozzle are similar, they may be depended upon to give satisfactory results for any diameter of front end now in use, or likely soon to come into use. Notwithstanding the short time which has elapsed since the first publication of the equations which have been given, they have already been extensively employed in service, with satisfactory results. The conditions which should be observed in their use is best shown by Figs. 142 and 143.

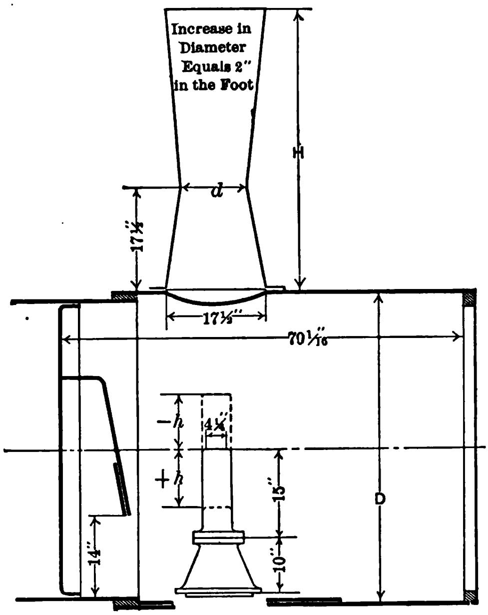


FIG. 142.

In this connection, also, it should be noted that it is not claimed that the plain stack and nozzle, as shown, will give better results than some other arrangement, but merely that when the plain stack and nozzle are used the equations will give the best relation of diameter to height which is obtainable. It is this question only that the experiments were designed to cover. Whether, for example, as a general proposition, the application of draft- or petticoat-pipes will improve the draft, or whether they will affect the relation of height and diameter of stack as already established, cannot be determined from the present work.

119. Unavoidable Loss in Draft with Deduction in Height of Stack.—The equations already presented, together with the tabulated statements based thereon, are assumed to give the best diameter for a stack of any given height. They cannot be depended upon to give results which will always be satisfactory under conditions which greatly limit or restrict the height. In general, the best draft obtainable from a short stack is inferior to that obtained from a longer stack. The most that can be done when the limit of height has been fixed is to

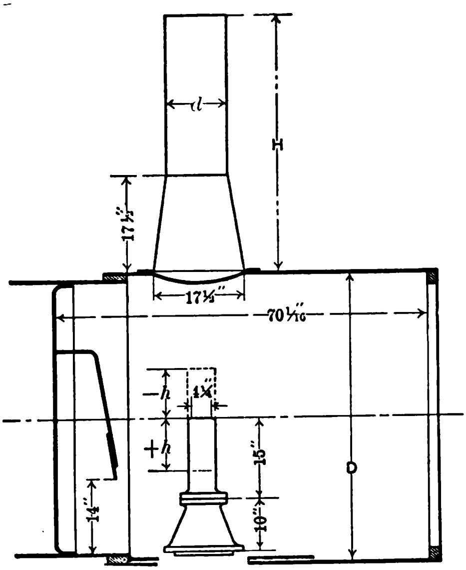


FIG. 143.

determine a diameter which, in combination with that height, will give best results. It is this, and this only, that the equations, and the tables based thereon, are assumed to do.

The rate at which the draft diminishes with each reduction in height of stack is indicated by Figs. 144 to 147, presenting results plotted in terms of draft and stack height. In explanation of these figures, it should be noted that the stack heights represented are: Base, 16½ inches; *A*, 26½ inches; *B*, 36½ inches; *C*, 46½ inches; and *D* 56½ inches. Fig. 144 gives results with the smallest tapered stack in combination with the lowest nozzle; Fig. 145, those for the largest tapered stack in combination with the lowest nozzle; and Figs. 146 and 147 those for the smallest and largest tapered stacks respectively, for a

series of tests at different cut-offs. In all of these figures it will be seen that there is a marked decrease in the draft obtained for each reduction

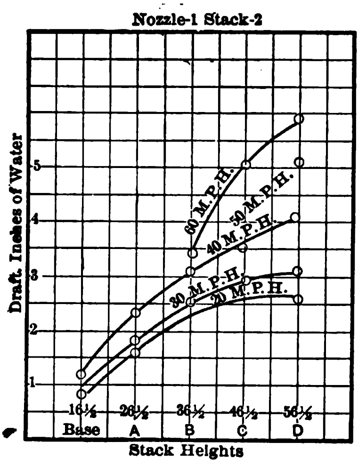


FIG. 144.—Small Tapered Stack.

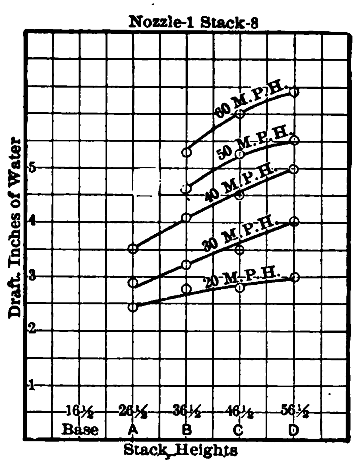


FIG. 145.—Large Tapered Stack.

in the height of stack. This is the more significant when it is considered that these diagrams represent only tapered stacks, and that the

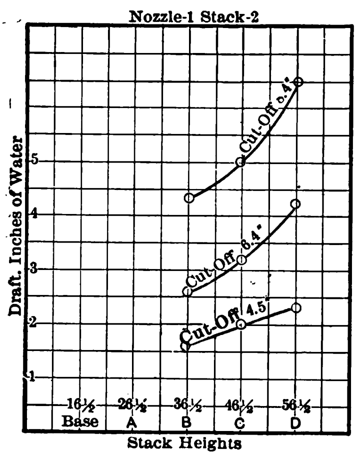


FIG. 146.—Small Tapered Stack.

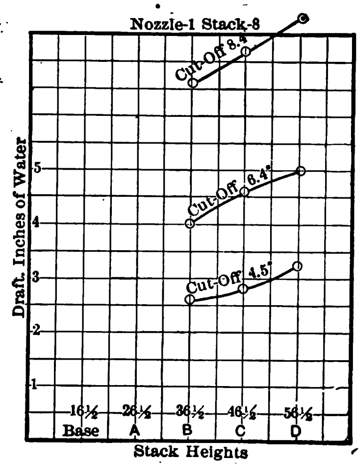


FIG. 147.—Large Tapered Stack.

diameter of the choke of such stacks is not required to be changed when the stack is varied in height. It is clear, therefore, that these diagrams confirm the statement already made to the effect that a

short stack, however well designed, must be inferior in its draft-producing qualities to a longer stack of good design. There is, in fact, nothing in the relation of diameter to height, or, so far as the investigation has proceeded, in the form of the stack itself, which can be accepted as a complete substitute for height. Again, it can be shown that this loss in draft with changes in height of stack does not depend upon the height of the nozzle. Thus, Table LVII. represents a summary of results in connection with all heights of the smallest tapered stack in combination with all heights of nozzle. The table shows that, whatever the height of the nozzle, the loss in draft under the conditions represented in passing from the *D*, or 56½-inch height, to the *A*, or 26½-inch height, is a fixed quantity, and under the conditions of the tests referred to is approximately represented by 1.6 inches of water. Thus, for the nozzle No. 1, it is 1.7 inches; for the nozzle No. 3, 1.4 inches; for the nozzle No. 5, 1.8 inches; for the nozzle No. 7, 1.6 inches.

TABLE LVII.

BEST DRAFT OBTAINABLE UNDER CONDITIONS OF CONSTANT SPEED AND CUT-OFF WITH DIFFERENT HEIGHTS OF STACK IN CONNECTION WITH DIFFERENT HEIGHTS OF EXHAUST-NOZZLE.

Nozzle.	1		2		3		4		5		6		7	
	Draft.	Difference.	Draft.	Difference.	Draft.	Difference.	Draft.	Difference.	Draft.	Difference.	Draft.	Difference.	Draft.	Difference.
Stack 2-D...	4.0	....	4.2	....	4.0	....	3.9	....	4.0	....	3.4	....	3.2	....
Stack 2-C...	3.5	.5	3.8	.4	3.7	.3	3.5	.4	3.5	.5	3.1	.3	2.7	.5
Stack 2-B...	3.1	.4	3.2	.6	3.3	.4	3.0	.5	2.9	.6	2.5	.6	2.2	.5
Stack 2-A...	2.3	.8	2.4	.8	2.6	.7	2.4	.6	2.2	.7	2.0	.5	1.6	.6
Total difference.....	....	1.7	....	1.8	....	1.4	....	1.5	....	1.8	....	1.4	...	1.6

It is difficult to find a quantitative measure for the loss of draft resulting from reduction in height of stack which will be of general application. The extent of such loss is, however, suggested by the slope of the lines in the figures already referred to, and by the differences in Table LVII. The general conclusion is, however, clear. It is apparent that the shorter stacks are far inferior to those of greater length, and that the maintenance of draft values in connection therewith must necessarily involve the application of additional mechanism, or an increase in the energy of the exhaust-jet. The study here de-

scribed does not suffice to indicate what should be the character of such additional mechanism nor the extent of the necessary increase in the energy of the exhaust-jet. It is, however, altogether possible that the adoption of some form of inside stack or of draft-pipes may make good the loss resulting from the diminished length of outside stack.

120. Relative Advantage of Straight and Tapered Stacks.—But two forms of stacks were employed in the experiments under discussion, one being perfectly cylindrical, and hence referred to as the straight stack; the other having the form best shown by Fig. 122, and generally referred to as the tapered stack. This tapered stack has its least diameter, or choke, at a point 16½ inches from the bottom, and increases in diameter uniformly above this point, the angle of the sides being the same for all stacks, and the divergence being at the rate of 2 inches in diameter for each foot in length. This divergence corresponds with that which was found best, as a result of the von Borries-Troske tests. The most noteworthy difference in results obtained from the two contours is to be found in the increased capacity of the tapered stack. Thus, No. 2 stack (9¾ inches diameter), which is the smallest of the tapered stacks, gives draft values which are two or three times as great as those obtained with No. 1, which is a straight stack of the same diameter. While the results from No. 2 are none of them sufficiently meritorious to warrant a place in the table of best results (Table LVII.), they give a close approach thereunto. The two smaller diameters of straight stacks Nos. 1 and 3 (9¾ and 11¾ inches diameter) are both far too small to yield results which are to be regarded as satisfactory, the capacity of these stacks being insufficient for the work expected of them. Speaking in rather general terms, it may be said that a tapered stack having a diameter of choke of 9¾ inches has as great a capacity as a cylindrical stack of 13¾ inches. Again, only the largest diameter of straight stack No. 7 (15¾ inches diameter) has a place in the table of best results, whereas all diameters of the tapered stack, excepting the least, find places in the table. It is of interest to note that with the low nozzle, tapered stack 6 *B* (least diameter, 13¾ inches; total height, 36½ inches) gives identical results with straight stack 7*C* (diameter, 15¾ inches; height, 46½ inches), the tapered stack being 10 nches lower and 2 inches less in diameter than the straight stack.

It has been stated that whereas for best results the diameter of the straight stack must change for every change in height, that of the tapered stack remains constant for all heights. In this connection it is

important to remember that the diameter of the tapered stacks referred to is the diameter of the choke (least diameter). As a matter of fact, if designs based on the equations which have been deduced were to be superimposed, they would show that while the tapered stack, as compared with the straight stack, is of lesser diameter at the choke, the diameter of its top would generally exceed that of the straight stack.

In general, it would seem that the tapered stack is less susceptible to minor changes of proportion, both of the stack itself and of the surrounding mechanism, than the straight stack. Thus, a variation of an inch or two in the diameter of the tapered stack affects the draft less than a similar change in the diameter of the straight stack. Again, the tapered stack is generally less affected by variations in the height of the nozzle, so that altogether it appears quite contrary to the expectation of the experimenter, that the use of the tapered stack gives a greater degree of flexibility in the design of the dependent parts. For these reasons the tapered form is altogether preferable to the straight form.

121. A Summary of Results.—The most important conclusions to be drawn from the experiments described may be stated as follows:

1. The jet acts upon the smoke-box gases in two ways; first, by frictional contact, it induces motion in them; and, second, it enfolds and entrains them.

2. The action of the jet upon the smoke-box gases is to draw them to itself, so that the flow within the front end is everywhere toward the jet.

3. The action of the jet is not dependent upon the impulses resulting from individual exhausts. Draft can be as well produced by a steady flow of steam as by the intermittent action of the exhaust.

4. Draft resulting from the action of the jet is nearly proportional to the weight of steam exhausted per unit of time. It does not depend upon the speed of the engine nor the cut-off of steam from the cylinders, except in so far as these affect the weight of steam exhausted.

5. The form of the jet is influenced by the dimensions of the channels through which it is made to pass. Under ordinary conditions, it does not fill the stack until near its top. If the diameter of the stack is changed, that of the jet will also change.

6. All portions of the smoke-box which are in front of the diaphragm have substantially the same pressure; and, consequently, a draft-gauge attached at any point may be depended upon to give a true reading.

7. The resistance which is offered to the forward movement of the air and gases between the ash-pan and the stack may be divided approximately into three equal parts which are: First, the grate and the coal upon the same; second, the tubes; and, third, the diaphragm. It is significant that the diaphragm is as much of an impediment to draft as the fire upon the grate.

8. The form and proportions of the stack for best results are not required to be changed when the operating conditions of the engine are changed; that is, a stack which is suitable for one speed is good for all speeds, and a stack that is suitable for one cut-off is good for all cut-offs. In future experiments of draft appliances, therefore, results obtained from a single speed and a single cut-off should be deemed satisfactory.

9. Other things remaining unchanged, the draft varies with the weight of steam exhausted per unit of time; if the number of pounds of steam exhausted per minute is doubled, the draft, as measured in inches of water, is doubled; if it is halved, the draft value is halved.

10. As regards the form of outside stacks, either straight or tapered may be used. From a designer's point of view, the tapered is the more flexible; that is, with the tapered stack, the draft is less affected by slight departures from standard dimensions. Incidental reasons, therefore, make the tapered form preferable. For best results the diameter of a given straight stack should be greater than the least diameter of a tapered stack for the same conditions.

The term "tapered stack" used in this and other paragraphs signifies a stack having its least diameter or choke $16\frac{1}{2}$ inches from the bottom, and a diameter above this point which increases at the rate of 2 inches for each foot in length.

11. In the case of outside stacks, either straight or tapered in form, the height is an important element. In general, the higher the stack the better will be the draft.

12. The diameter of any stack designed for best results is affected by the height of the exhaust-nozzle. As the nozzle is raised the diameter of the stack must be reduced, and as the nozzle is lowered the diameter of the stack must be increased.

13. The diameter of a straight stack designed for best results is affected by the height of the stack. As the stack height is increased, the diameter also must be increased.

14. The diameter of a tapered stack designed for best results, as

measured at the choke, is not required to be changed when tne stack height is changed.

15. The precise relation between the diameter of front end and the diameter and height of stack for best results is expressed by equations as follows:

For straight stacks:

When the exhaust-nozzle is below the center line of the boiler

$$d=(.246+.00123H)D+.19h. \quad (8)$$

When the exhaust-nozzle is above the center line of the boiler

$$d=(.246+.00123H)D-.19h. \quad (9)$$

When the exhaust-nozzle is on the center line h is equal to zero and the last term disappears, and there remains

$$d=(.246+.00123H)D. \quad (10)$$

For tapered stacks:

When the nozzle is below the center line of the boiler

$$d=.25D+.16h. \quad (11)$$

When the nozzle is above the center line of the boiler

$$d=.25D-.16h. \quad (12)$$

When the nozzle is on the center line of the boiler h becomes zero. and

$$d=.25D. \quad (13)$$

In all of these equations d is the diameter of the stack in inches For tapered stacks it is the least diameter or diameter of choke. H is the height of stack in inches, and for maximum efficiency should always be given as large a value as conditions will admit. D is the diameter of the front end of the boiler in inches, and h the distance between the center line of the boiler and the top of the exhaust-tip.

122. Later Experiments.—The preceding paragraphs present an account of researches upon the locomotive front end up to and including the year 1902. The work then completed left unsettled several important questions, namely, the best proportions for and the

value of inside stacks, false tops within the front end, and draft-pipes. The equations which had resulted from the previous work were assumed to have been of general application, but their truth when applied to locomotives having short stacks upon boilers of large diameter had not been confirmed by direct experiment. Upon the completion of the work already described, therefore, the American Railway Master Mechanics' Association appointed a committee to co-operate with the American Engineer and Purdue University in doing such work as still remained in determining the proper proportions of the several details making up the locomotive front end, and to raise by subscription among railroad companies the funds needed to meet the expense thereof. The following is abstracted from the final report of this committee: *

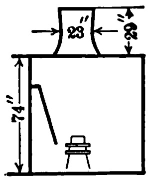
The tests were made in connection with New York Central locomotive No. 3929 mounted upon the Purdue testing plant. The locomotive was of the Atlantic type having 21″×26″ cylinders and 72″ drivers. The inside diameter of the smoke-box was 74″, and the height of its normal stack 29″. The methods followed were similar to those already described.

The tests of outside stacks involved a height of 29 inches, this being the maximum practicable for road conditions upon the locomotive under test. Stacks of this height were supplied in diameters ranging from 15 to 25 inches by 2-inch steps. In these tests no draft-pipes or netting were employed in the front end; the diaphragm and exhaust-pipe were the only details present. Under these conditions, with a 29-inch height, the best diameter was found to be 23 inches, though this was not much better than that of 21 inches. The exact arrangement of equipment for the best results is shown by Fig. 148. The notation under this figure and under those which immediately follow gives the draft obtained with a constant back pressure of 3.5 pounds. It will hereafter appear that there are better arrangements than that shown by Fig. 148. The point which is proven is that, assuming a plain outside stack 29 inches high to be used, its diameter for best results is 23 inches, as given.

The experiments included inside stacks of four different diameters ranging from 15 to 21 inches, a constant outside height of

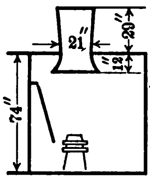
* The report in full will be found in the Proceedings of the Master Mechanics' Association for June, 1906.

29 inches, and a penetration into the smoke-box of 12, 24, and 36 inches, respectively. The best proportions for this form of stack are shown by Fig. 149 accompanying. Its diameter is 21 inches and its penetration into the smoke-box is 12 inches. Results of nearly the same value were, however, obtained with stacks of smaller diameter having greater penetration. From values obtained it appears that as the degree of penetration increases the diameter of stack should be reduced. The effect is, in fact, of the same nature and degree as that which results from raising the exhaust-tip. It is noteworthy also that these values for the plain inside stack are not materially better than those for the plain outside stack, a fact which was formulated as a conclusion resulting from the work of 1902.



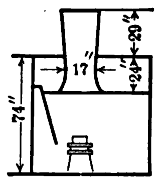
DRAFT, 4.54

FIG. 148.



DRAFT, 4.71

FIG. 149.



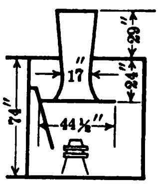
DRAFT, 5.06

FIG. 150.

It had been planned to fit the front end with three different false tops located at 12, 24, and 36 inches, respectively, from the top of smoke-box, but the presence of the steam-pipes made it difficult to fit the 12-inch top, and as a consequence only those of 25 and 36 inches drop were experimented upon. In each case stacks of different diameters were used, the outside height being always 29 inches. The best results were obtained with a stack 17 inches in diameter having a penetration of 24 inches, all as shown by Fig. 150. The draft resulting therefrom, when the back pressure was 3.5 pounds, was 5.06. In all cases with the false top the 17-inch stacks gave best results. A comparison of results with those obtained from a plain outside stack and from a plain inside stack shows material improvement in draft values.

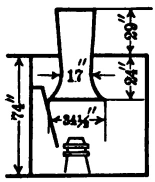
The false top necessarily interferes with free access into the front end, which fact makes it desirable that a way be found in which to secure the results derived from it by means which are more simple. It was suggested that experiments be made to determine the effect upon the plain inside stack of an annular ring or flange which might

be considered as representing a portion of the false top. Responding to this suggestion rings of two diameters were used on 17-inch and 19-inch stacks having a penetration of 24 inches. It was found that the proportions shown by Fig. 151 gave substantially the same results as those obtained with the best arrangement of false top. Believing that the results thus obtained pointed to the desirability of having a broader curve at the base of the stack and that when the proper proportions were understood the best results would be obtained from such a curved surface, the 17-inch stack was fitted with a bell to which, for purposes of experimentation, flanges of various widths were afterward added, with the result that those proportions which appear in Fig. 152 proved most satisfactory. The best draft with the false top was 5.06, with the ring 5.05, and with the bell 4.98—that is, these three arrangements are practically on an equality



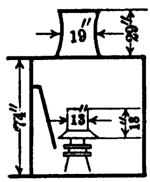
DRAFT, 5.05

FIG. 151.



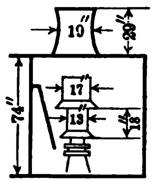
DRAFT, 4.98

FIG. 152.



DRAFT, 4.55

FIG. 153.



DRAFT, 4.40

FIG. 154.

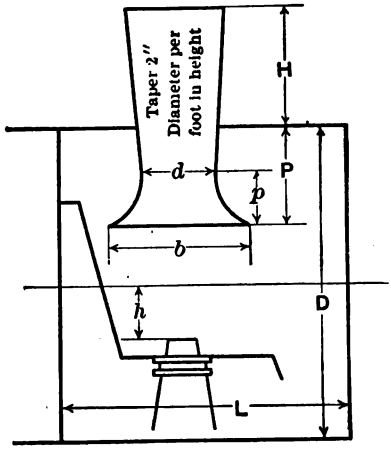
No other arrangements were experimented upon which gave higher draft values than these.

Draft-pipes of various diameters, adjusted to many different vertical positions, were tested in connection with plain stacks of the several diameters available. The elaboration of this phase of the work was very extensive. It was found that for best results the presence of a draft-pipe requires a smaller stack than would be used without it, but that no possible combination of single draft-pipe and stack could be found which would give a better draft than could be obtained by the use of a properly proportioned stack without the draft-pipe. While the presence of a draft-pipe will improve the draft when the stack is small, it will not do so when the stack is sufficiently large to serve without it. The best proportion and adjustment of single draft-pipe and stack are shown by Fig. 153.

Double draft-pipes of various diameters and lengths, and having many different positions within the front ends, all in combination with stacks of different diameters, were included in the experiments

with results which justify a conclusion similar to that reached with reference to single draft-pipes. Double draft-pipes make a small stack workable. They cannot serve to give a draft equal to that which may be obtained without them, provided the plain stack is suitably proportioned. The arrangements and proportions giving the best results are those shown by Fig. 154.

A suggestion as to a standard front end is presented as Fig. 155,



BEST ARRANGEMENT OF FRONT END.

FIG. 155.

which, with the following equations referring thereto, may be accepted as a summary of the conclusions to be drawn from all experiments made.

For best results make H and h as great as practicable. Also make

$$d = .21D + .16h, \qquad (14)$$

$$b = 2d \text{ or } .5D, \qquad (15)$$

$$P = .32D, \qquad (16)$$

$$p = .22D. \qquad (17)$$

CHAPTER XII.

SUPERHEATING IN THE SMOKE-BOX.*

123. **An Experimental Determination.** — The arrangement of steam-piping on an American locomotive constitutes one of the many ingenious features which have so long served to perpetuate the general characteristics of the early machine. From the time the steam leaves the dome of the boiler until it comes within the influence of the cylinders, there is no possible chance by which it can lose any of its heat, and during a portion of its course it actually gains heat. The arrangement as a system of piping is perfect. To a limited degree the pipe also serves as one element of a superheater, the smoke-box constituting the other. The extent of the drying or superheating effect upon the steam, which results from having the T head and the two branch pipes within the smoke-box, has long been an open question. Some foreign designers, evidently regarding the thermal gain resulting from such an arrangement as of less consequence than accessibility of parts, have carried the pipe connections outside of the smoke-box; while in this country at least one designer, not satisfied with following the usual practice, has carefully planned the form of his piping to facilitate the transmission of heat through its walls. To throw some light on the extent of the drying or the superheating effect upon the steam while passing these pipes in the smoke-box, a careful test was made upon the Purdue University experimental locomotive.

In preparation for the experiment a thermometer was inserted in the T head at A (Fig. 156), another in the middle of one branch of the steam-pipe at B, and a third in the saddle close to the valve-box at C. The locomotive was then run under load with the reverse-lever forward and the throttle only partially open, the drop in pressure from the boiler to the pipe caused by throttling being sufficient to superheat all of the steam as it expanded from the pressure of the boiler to that of

* The substance of this chapter was contributed to the *Railway Review*, July 28, 1894.

the pipe. The extent of the superheating of steam at any given pressure is determined from its temperature alone. If, in the case under consideration, the steam neither received nor gave up heat in its passage of the pipe, all three thermometers would show the same temperature. A difference in the reading of the thermometers, therefore, must indicate a transmission of heat.

The conditions of each test were maintained for a half-hour before any observations were made; the thermometers were then read and

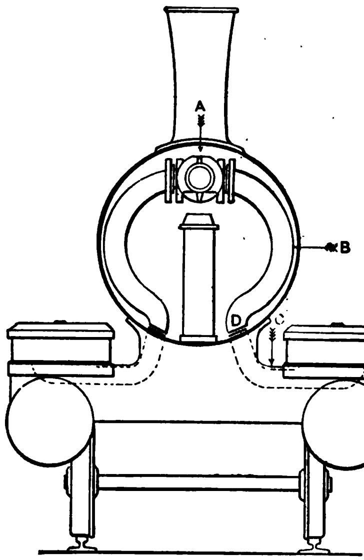


Fig. 156.

other observations taken simultaneously, at five-minute intervals, for a second half-hour, there being no difficulty in maintaining constant conditions. As affecting the reliability of results it should be said that the thermometers used had a range of from 100° to 200° C., and read to fifths of a degree. They were inserted in long tubes, and at *A* and *B* these tubes were protected by allowing steam under the pressure of the pipe to flow past them. Fig. 157 shows the arrangement used at *A* for obtaining the temperature of the steam in the T head. A similar arrangement was used at *B*. Before the tests were made the thermometers were carefully compared by exposing them, while in the identical tubes used on the locomotive, to saturated steam of about the same temperature as that recorded in the experiment. The read-

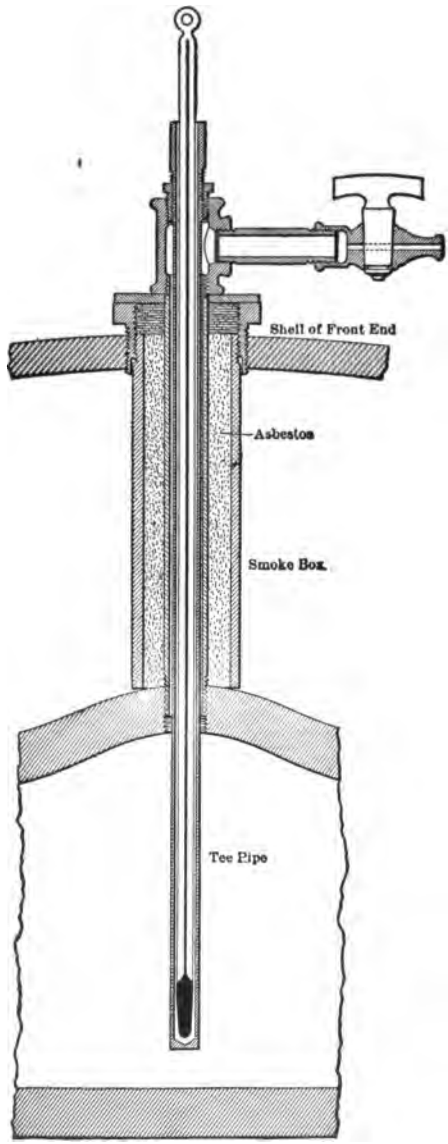


FIG. 157.

ing of one was accepted as standard, and the errors of the other two determined. Since all results depend upon differences of temperature, a slight error in the actual temperature was not considered important. Only the corrected readings are given.

The following is a summary of results for one of the tests:

Pressure of steam in boiler by gauge	130 lbs.
Pressure of steam in steam-pipe by gauge	70 lbs.
Temperature of saturated steam at pressure of steam-pipe	315.68° F.
Observed temperature of steam at *A*, Fig. 156 (at T head)	335.30° F.
Observed temperature of steam at *B*, Fig. 156 (at middle of steam-pipe)	339.65° F.
Observed temperature of steam at *C*, Fig. 156 (at saddle)	327.81° F.
Observed temperature of gases in smoke-box	700.00° F.
Increase of smoke-box temperature over temperature of steam in the boiler	345.00° F.
Approximate distance from *A* to *D*, measured along center line of pipe	56 in.
Size of branch pipe, inside	3×6 in.
Thickness of walls	$\frac{3}{8}$ in.
Approximate time occupied by the steam in passing from *A* to *C*	0.1 sec.

It will be seen that the temperature of the steam was increased 4.4° in passing from *A* to *B*, which is equivalent to a gain of 8.8° in passing through the whole length of the branch pipe. The transfer of a quantity of heat represented by an increase of 8.8° in the temperature of superheated steam would affect moist steam by increasing its dryness about 0.5 of 1 per cent. It is believed that for the engine experimented upon, this approaches the maximum benefit which, from a thermal point of view, is to be derived from having the pipes inside the smoke-box.

The conditions of the test were varied, and all the work repeated several times with the same general result. When a larger quantity of steam passed the pipe, the smoke-box temperature and the total heat transmitted increased, but the amount of heat transmitted per pound of steam was not materially changed. The figures given are from the test which showed the greatest heating effect.

Enlarging the pipes within the smoke-box would have a pronounced effect in increasing the action herein considered, since it would both add to the extent of heating surface and also lengthen the time occupied by the steam in passing the same; but as a practical matter a limit to such enlargement for simple engines is soon reached. It therefore seems unreasonable to expect the steam to be dried to any consider-

able extent in its passage of the smoke-box. Nevertheless, there is some gain, and a little gain is far better than a little loss.

It is likely that a two-cylinder compound may reap some material advantage from the heat transmitted to its receiver, for in this case the heating surface may be quite extensive, and the movement of the steam through the receiver comparatively slow. Moreover, the receiver pressure being lower than the pipe pressure of a simple engine, there is a greater difference of temperature between the smoke-box gases and the steam.

An interesting result of the test is found in the fact that the thermometer in the saddle at C indicated a temperature 7.5° lower than the temperature in the T head at A, and 16.3° lower than the presumable temperature at the lower end of the steam-pipe at D, so that, from the T head to the cylinder the steam does not gain, but actually loses heat. This effect is to be accounted for in the fact that the mean temperature within the cylinders is much lower than the temperature of the incoming steam, and that this, combined with the effect of radiation from the saddle, operates to lower the temperature of the iron which surrounds the steam in its course through the saddle. It is certainly clear that the heat given the steam by the smoke-box is soon taken away again by the saddle, but how much of this cooling effect of the saddle is due to radiation is not shown. Experience with stationary engines which have no saddles, however, forbids our charging it all to radiation.

IV. THE ENGINES.

CHAPTER XIII.

INDICATOR WORK.

124. Concerning Indicator Work.—From the beginning of the investigations of the locomotive laboratory the importance of giving close attention to the work of the indicators employed upon the locomotive cylinders was recognized. A good indicator, if well cared for and intelligently operated, can be depended upon to give an accurate record of the changing pressure within the cylinder, but the same instrument, if neglected or if carelessly manipulated, may give results which are worthless. In the work of the locomotive laboratory it was sought not only to have the indicator itself in good condition, but to have its action such as to insure a high degree of accuracy in its use. The reducing mechanism for transmitting the motion of the piston and of the indicator-drum is shown by Fig. 158. The motion of the cross-head is transmitted to the pendulum *A* through a pin working in a slot of the lever. Another slot in this lever, near its fulcrum, receives and serves to drive a pin mounting a roller fixed to the metal bar *B*, which slides in the guides *CC*. The pin driving the lever *A* and the pin upon the bar *B*, which is driven by this lever, can only move in lines parallel with the motion of the cross-head. As, in response to the motion of the cross-head, the lever *A* assumes its various angular positions, the leverage with which the driving-pin acts upon it is constantly changing, but similar changes are taking place in the leverage with which the lever *A* acts upon the pin which is attached to the bar *B*, hence the ratio of the two arms remains constant, or would do so if the pin and the rollers which they bear were infinitely small. The actual error arises from the measurable diameter of these pins, and is so slight as to be of no practical consequence. The presence of the bar *B*, having a motion precisely similar with that of the piston and in line with the drum-sheave of the indicator, permits the use of a short cord connection in driving the latter. Neatly made

blocks of wood clamped to the bar *B* serve as points of attachment for the indicator-cards, separate points being employed for each indicator. The connection of the steam end of the indicator with the cylinder also was designed to be as close as possible. This consists of a 3-inch length of ¾-inch pipe and one ¾-inch by ½-inch elbow, the smaller end of which receives the indicator-cock.

The cards obtained by means of this equipment were at once seen to be strikingly different from those obtained from locomotives upon the road, where a considerable length of pipe between the cylinder and indicator appears to be absolutely necessary. The cards from the

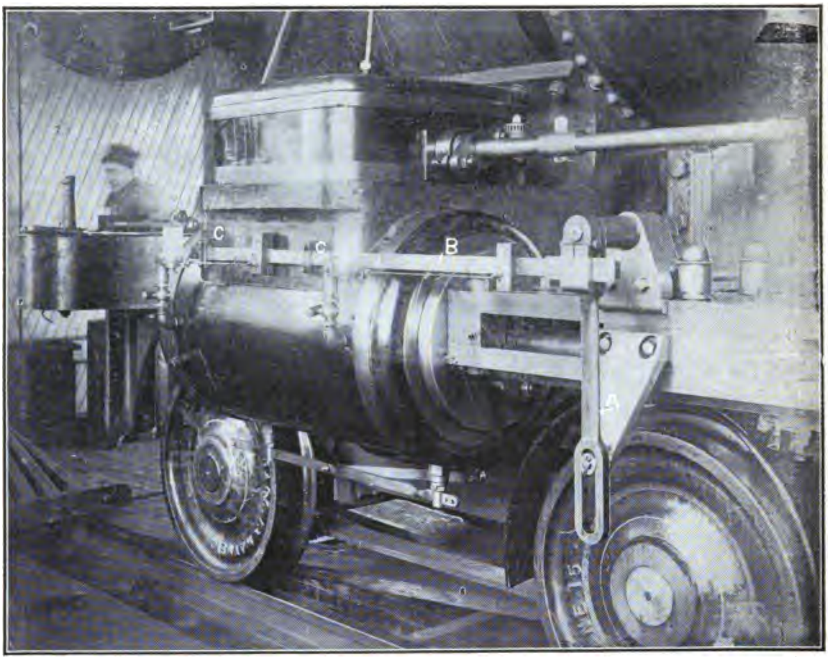


Fig. 158.—Indicator Rigging.

laboratory were smoother in their general outline, and the events of the stroke were much more clearly marked upon them. While these differences are entirely creditable to the laboratory, the fact that they existed gave rise to numerous inquiries. Different parties were found to be interested in comparing cards from the laboratory with those obtained from road tests, with no satisfactory results. As an aid to the interpretation of cards obtained upon the road, therefore, it was determined to make a study of the effect of an indicator-pipe upon the

form of the cards. The results of this study are of sufficient interest to warrant some reference to them in this connection, though the fact should be borne in mind that all indicator work, underlying the data of succeeding pages, was obtained under the favorable conditions set forth by Fig. 158.

125. **The Effect upon the Diagrams of Long Pipe Connections for Steam-engine Indicators.**—Experiments were first undertaken upon the locomotive which was equipped with two indicators, as shown by Fig. 159. One of these, hereafter referred to as the cylinder-indicator, was mounted upon the usual close connection; the other, hereafter referred to as the pipe-indicator, was mounted at the end of such a length of ¾-inch pipe as was necessary to carry the indicator to the

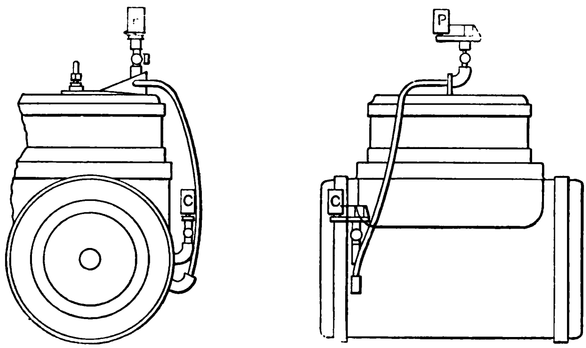


FIG. 159.

top of the steam-chest, as must ordinarily be done in conducting road tests. The pipe was approximately 3½ feet long, but it was bent to form smooth curves, and was carefully covered throughout its entire length. By these means it was thought that the interference resulting from the presence of the pipe would be reduced to a minimum. It was sought not to exaggerate differences in the results occurring through the presence of the pipe, but rather to reduce such differences to minimum values.

In carrying out the experiments cards were taken simultaneously from both indicators at various speeds. The indicators were then reversed in their positions, and the observations repeated. The averages of results thus obtained were employed as final values. A comparison of results discloses the fact that the events of the stroke (cut-off, release, and beginning of compression), as recorded by the pipe-indicator, were all later than similar events as recorded by the cylinder-indicator. The cards from the pipe-indicator made all changes of pressure appear more

gradual than those from the cylinder-indicator, and the area of the card from the pipe-indicator was greater. Fig. 160 presents two cards taken simultaneously from the indicators in question at a speed of 50 miles an hour, and Table LVIII. presents the differences in power as obtained from the two indicators for all speeds between 25 and 35 miles an hour.

TABLE LVIII.

ERRORS IN M.E.P. CAUSED BY PIPE CONNECTION SHOWN IN FIG. 159.

Speed, Miles per Hour.	Speed, Revolutions per Minute.	Excess of Power shown by Pipe-indicator as Compared with that shown by Cylinder-indicator in Per Cent.
25	134	1.5
30	161	2.1
35	188	2.9
40	215	4.9
45	242	8.4
50	269	14.0
55	296	17.2

126. Experiments upon a Stationary Engine.—The significance of the results described in the preceding paragraph was such that a

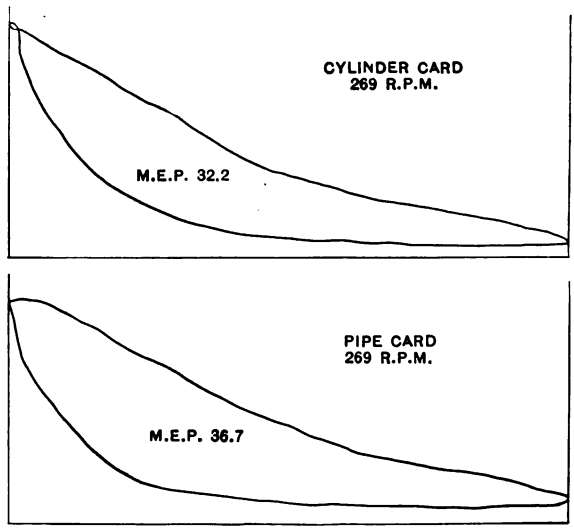


FIG. 160.

more elaborate process of experimentation was entered upon. As the locomotive was not well adapted to the purposes in hand, the experiments were transferred to a Buckeye engine having a cylinder 7¾

inches in diameter by 15 inches stroke. Since a description of these later experiments will serve to enforce conclusions which may be drawn from those already described, some reference to them in this connection will be of interest.*

The power of this engine was absorbed by an automatic friction-brake, by means of which a very constant load was obtained. The head end of the engine cylinder was tapped with two holes (*a* and *b*, Fig. 161), both in the same cross-section, and hence equally exposed to the action of the steam in this end of the cylinder. One of these holes (*a*) was made to serve for the indicator *A*, the cock of which was placed as close to the cylinder as possible. The hole *b* was made to receive one end of a U-shaped pipe, the other end of which entered a coupling fixed in the angle-plate *c*. The cock of a second indicator, *B*, was screwed to this coupling. A single system of levers supplied the drum motion for both indicators. The pipe fittings were all half-inch. A right-and-left coupling at *d* allowed the U-shaped section, *dfb*, to be removed at will, and replaced by a similar section of different length. Pipe lengths of 5, 10, and 15 feet were used, length being measured from the outside of the cylinder wall to the end of the coupling under the cock of the pipe-indicator. The pipe and fittings were covered first with a wrapping of asbestos board, next with three-eights of an inch of hair felt, and finally with an outside wrapping of cloth. It is to be noted that the bend in the pipe at *f* is easy, and that there is a continual rise in the pipe in its course from the cylinder to the indicator. Both indicators were always well warmed before cards were taken. A gauge between the throttle and the valve-box was useful as an aid in securing constant pressure within the latter. In the tests herein described, however, the boiler pressure was kept constant, as nearly as possible, and the throttle was generally fully open.

A pair of new Crosby indicators was set apart for this work, and while it will be shown that the value of the comparisons which were undertaken is not dependent upon a high degree of individual accuracy in the indicators, these instruments, when calibrated under steam, gave results which were nearly identical.

The results, which are presented in the form of diagrams (Figs. 162 to 171), were obtained in the following manner:

* An account of results obtained from experiments in connection with the Buckeye engine was first published as a paper before the American Society of Mechanical Engineers, "The Effect upon the Diagrams of Long Pipe Connections for Steam-engine Indicators," Proceedings of the Society, May, 1896.

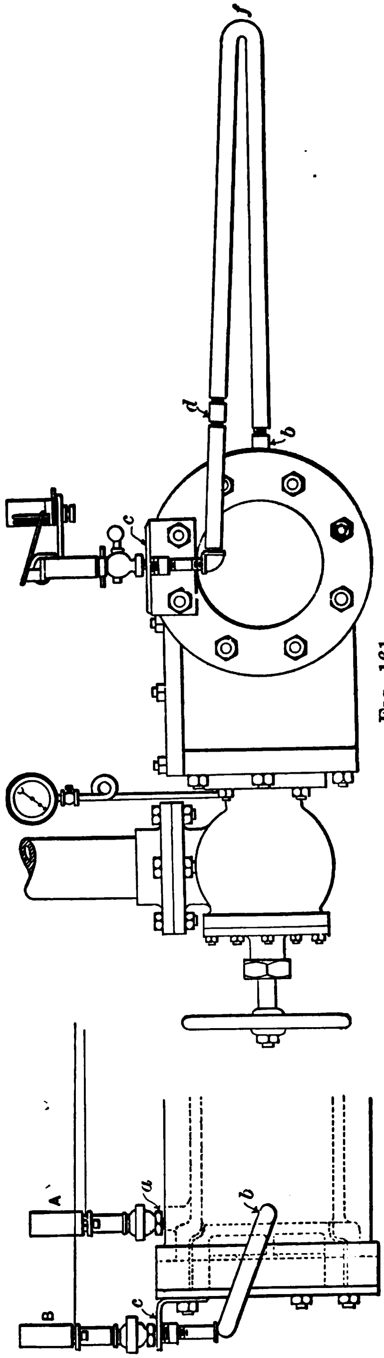


FIG. 161.

The engine having been run for a considerable period, and the desired conditions as to pressure, speed, and cut-off having been obtained, cards were taken simultaneously from the cylinder- and the pipe-indicator. Two pairs of cards (i.e., two from cylinder and two from pipe) were thus taken as rapidly as convenient, after which the position of the indicators was reversed, and the work repeated. There were thus obtained four cylinder-cards and four pipe-cards, one-half of each set having been made by one of the indicators, and one-half by the other. Next, by the use of closely drawn ordinates the eight cylinder-cards were averaged and combined in the form of a single card, and the eight pipe-cards were in the same way combined to form a single pipe-card. The two typical cards thus obtained, superimposed as in Fig. 162, constituted the record of the test. This process was repeated for each of the several conditions under which tests were made. It is proper to add that the accuracy of the indicators used, and the constancy of the conditions maintained, were such as to make each card almost, if not quite, the exact duplicate of the representative of its set.

The diagrams presented are full sized, the spring for all being sixty pounds.

127. **Different Lengths of Pipe.**—The effects produced by the use of pipes between the indicator and the engine cylinder, of five, ten, and fifteen feet in length, are shown in Figs. 162, 163, and 164 respectively, the speed, steam pressure, and cut-off being constant. As noted upon the figures, the full outline represents the cylinder-card, and the dotted outline the pipe-card.

It would seem that, under the conditions stated, the form of cylinder-cards in the figures referred to should be nearly the same, whereas the figures show them to vary considerably. It will be well to omit, for the present, all discussion concerning the causes of these differences, and to accept the cylinder-card in each case as representing the true conditions within the cylinder.

By reference first to Fig. 162 it will be seen that the effect of a five-foot pipe is to make the indicator attached to it a little tardy in its action. Thus, during exhaust, when for a considerable interval of time the change of pressure to be recorded is slight, the lines from the two indicators agree, but during the compression which follows the loss of sensitiveness in the pipe-indicator is made evident by its giving a line which falls below the corresponding line traced by the cylinder-indicator. Similarly, during admission there is an approximate agreement, while during the expansion which follows, the lagging of the

pipe-indicator results in a line which is higher than the expansion line given by the cylinder-indicator. As a result of this lagging in the

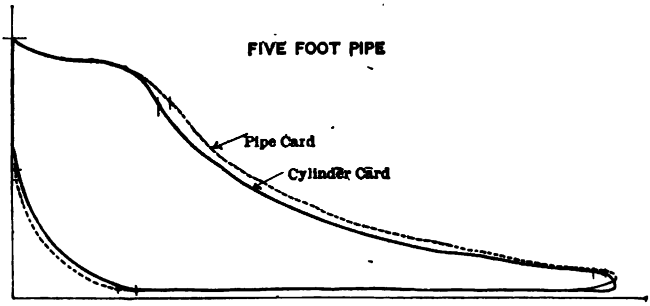


FIG. 162.

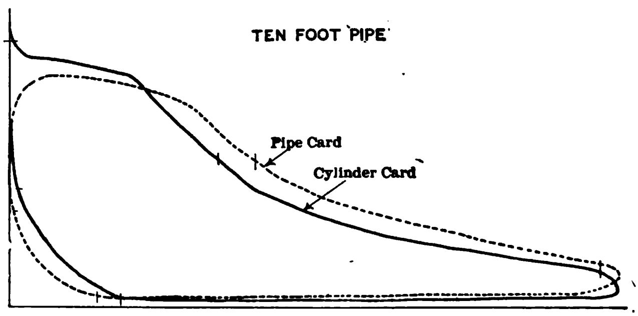


FIG. 163.

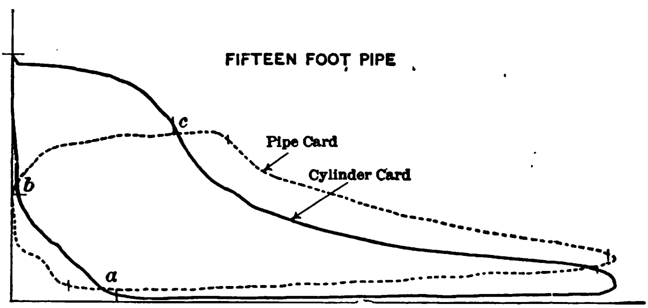


FIG. 164.

NOTE.—The speed (200 revolutions per minute), the steam pressure (80 pounds), and the cut-off (approximately ¼ stroke) were constant for all diagrams on this page.

action of the pipe-indicator, its card is in error in the location and curvature of the expansion and compression curves; also in the location of the events of the stroke, and in the area which it presents. The

speed at which these errors are shown to occur is moderate (200 revolutions), and the length of pipe attached to the indicator is not greater than is often used.

The general effect of a ten-foot length of pipe (Fig. 163) is the same as that of the shorter length, but the lagging action due to the pipe is more pronounced, and all errors are proportionately greater. In this case, also, the admission and exhaust lines fail to agree, the total range of pressure recorded upon the cards being less than the range existing in the cylinder.

A still further addition to the length of the pipe brings changes (Fig. 164) into the form of the pipe-card diagram which, while entirely in harmony with those already discussed, are of such magnitude that the form of the card loses some of its characteristic features. The admission and expânsion lines are lower and the exhaust line is higher than are the corresponding lines for the true card. Reference to Table LIX. wills how that while cards from pipes of five and ten feet in length present an area greater than that of the true card, the card in question (Fig. 164) from a fifteen-foot length of pipe makes the area less.

A comparison of the pipe-cards, Figs. 162, 163, and 164, makes it evident that a pipe of suitable length would result in a diagram somewhat similar in form to that shown by Fig. 165 and a pipe still longer would give a card which would be represented by a single line, as *AB*, Fig. 165.

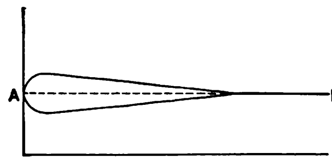


FIG. 165.

Various numerical results from Figs. 162, 163, and 164 are exhibited in Table LIX.

It is true that the lengths of some of the pipes experimented with are excessive as compared with those commonly used for the connection of indicators, but this fact does not deprive the results of their significance. If pipes of fifteen, ten, and five feet in length will produce the effects shown by Figs. 162, 163, and 164 respectively, it is but reasonable to suppose that pipes of less than five feet in length will produce some effect. And, since the effect of a five-foot pipe is considerable, this length must be greatly reduced before the effect ceases to be measurable.

It will be shown later that differences of speed have less effect than would be supposed in modifying the form of the pipe-card; that, even with a speed as low as 100 revolutions per minute, the effect of the pipe is strikingly apparent. It will be shown, also, that the point of cut-off chosen for the whole series now under consideration (Figs. 162, 163, and 164) is not especially favorable for showing the modifying effect of the pipe. These considerations, together with the fact that indicator-pipes of three and four feet in length are not uncommon, all serve to emphasize the practical value of the effects noted.

128. The Form of the Cylinder Diagrams.—It may be well at this point to consider the causes tending to change the form of the cylinder diagrams as they appear in the different figures (Figs. 162, 163, and 164), and to consider the evidence which justifies their acceptance as true diagrams. A study of the diagrams and data will make it evident that the differences are due wholly to a change in the length of the pipe. This single change, however, introduces an incidental change (1) in the clearance, (2) in the extent of surface exposed to the action of the steam, and (3) in the velocity of flow in and out of the pipe at the point of its connection with the cylinder.

The effect produced by the several pipes upon the clearance of the engine is given below:

Cylinder and port clearance, per cent of piston displacement	4.08
Clearance due to 5-ft. pipe, " " " " "	2.69
" " " 10-ft. " " " " " "	5.14
" " " 15-ft. " " " " " "	7.84
Total clearance with 5-ft. pipe in place, per cent of piston displacement	6.77
" " " 10-ft. " " " " " " " " "	9.22
" " " 15-ft. " " " " " " " " "	11.92

The area of surface bounding the clearance space was affected by the pipes as follows:

Surface bounding clearance space, no pipe attached	131.4 sq. in.
" " " " when 5-ft. pipe was in place	250.2 " "
" " " " " 10-ft. " " " "	366.1 " "
" " " " " 15-ft. " " " "	486.0 " "

Increased clearance would lower the pressure at the end of compression, and would change the curvature of the compression line but it would not make the compression line as it appears in Fig. 164.

The larger exposed surface would increase the effect due to the interchange of heat between the steam and the walls inclosing it. If it be assumed that, during the early stages of compression, this inter-

change results in reëvaporation, and during the later stages in condensation, the sum total of the effect would be in line with that recorded. Such an assumption is reasonable, and such an action may in part account for the change under discussion, but its extent is not likely to be as great as the indicator has recorded.

By far the most active agent tending to reduce the curved compression line of Fig. 162 to the straight line of Fig. 164 is the movement of steam in and out of the mouth of the pipe. Thus, when compression begins in the cylinder, the pressure at the end of the pipe is greater than that in the cylinder (see Fig. 164), and steam must flow from the pipe to the cylinder. This current of steam entering the cylinder just when the mixture of steam and water already there is undergoing the early stages of compression helps to augment the cylinder pressure, and to carry the early part of the compression line higher than it would otherwise go. As the process of compression goes on the current in the pipe is reversed, and the cylinder supplies steam to the pipe, thus causing the curve for this portion of the event to fall lower than it otherwise would. Increased curvature during the early stages and diminished curvature during the later stages result in a line which is approximately straight (Fig. 164).

Similar reasoning will account for the rapid drop in pressure after cut-off (cylinder-card, Fig. 164). At the instant of cut-off the cylinder is supplying steam to the pipe. The flow is rapid, and the kinetic energy of the steam causes it to pile up in the pipe; and, although as the stroke advances this pressure is constantly decreasing, the pipe continues throughout expansion to hold a higher pressure than that contained by the cylinder.

It is obvious that the pressure is not the same at the two ends of the pipe, except for points indicated by the crossing of the lines, as at *a*, *b*, and *c* (Fig. 164), and that the difference of pressure shown at other points is quite sufficient to account for the pronounced change in the cylinder diagram when pipes of different lengths are used. The existence of these differences in the form of the cylinder diagram does not in any way affect the results which are here presented. All cylinder-cards may be accepted as true, and the fact that they are not all alike does not diminish their value, but rather emphasizes the importance of this whole subject.

129. The Effect of the Pipe at Different Speeds.—The effects thus far discussed are those recorded for a constant speed of 200 revolutions per minute. In considering to what extent changes of speed will

modify these results, reference should be made to Figs. 166, 167, and 168, which give a series of results involving a ten-foot pipe for which all condi-

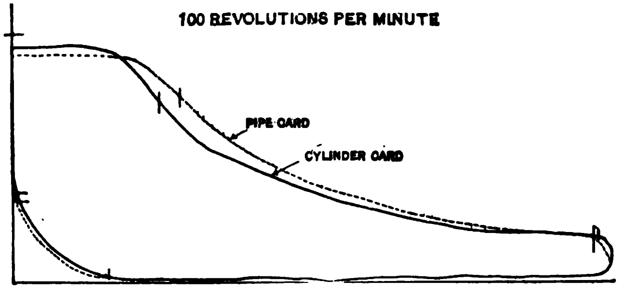


FIG. 166.

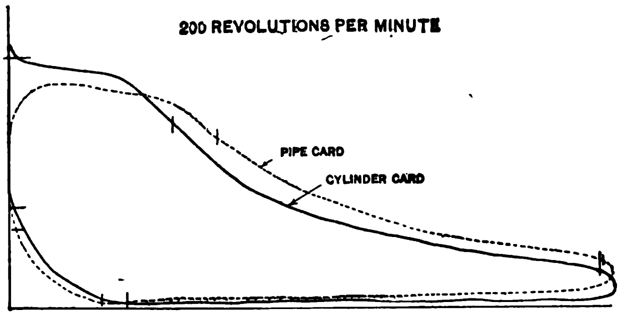


FIG. 167.

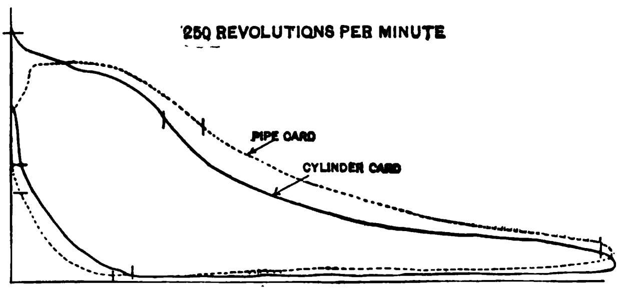


FIG. 168.

NOTE. The steam pressure (80 pounds), the length of pipe (10 feet) and the cut-off (approximately $\frac{1}{4}$ stroke), were constant for all diagrams on this page.

tions were constant except that of speed. Numerical comparisons may be made from Table LIX. It will be seen that increase of speed produces modifications in the form of the pipe diagrams which, in kind,

are similar to those produced at constant speed by increasing the length of the pipe, but these changes are not great. For example, increasing the speed from 100 to 200 revolutions per minute (Figs. 166 and 167) produces less change than increasing the length of the pipe from five to ten feet (Figs. 162 and 163). The fact that an engine runs slowly, therefore, does not seem to justify the use of an indicator at the end of a considerable length of pipe. Slow running reduces the error; it cannot be depended upon to eliminate it entirely.

130. The Effect of the Pipe at Different Cut-offs.—The relative effect of the pipe when the cut-off is changed, other conditions being constant, is shown by Figs. 169, 170, and 171, and numerically by Table LIX. It will be seen that the differences of pressure recorded during expansion by the two indicators (pipe and cylinder) are approximately the same for all cut-offs, but the relative effect of these differences upon the area of the diagram is most pronounced upon the smallest, or shortest, cut-off card. The fact that in Fig. 171 the steam line on the pipe-card rises, while that of the cylinder-card declines, constitutes a good illustration of the slowness with which the pressure in the pipe responds to that in the cylinder.

TABLE LIX.

THE EFFECT OF A PIPE ON THE FORM OF INDICATOR DIAGRAMS.

Test Number.	Apparent Cut-off. Excess shown by Pipe Diagram.		Pressures. Excess (+) or Deficiency (−) shown by Pipe Diagram.								Steam Consumption per Horse-power per Hour. Excess (+) or Deficiency (−) as shown by Pipe Diagram.	
			At Cut-off.		At Release.		At the Beginning of Compression.		M.E.P.			
	Per Cent of Stroke.	Per Cent. of Cyl. Cut-off.	Pounds.	Per Cent.	Pounds.	Per Cent.	Pounds.	Per Cent.	Pounds.	Per Cent.	Pounds.	Per Cent
1	3.0	11.5	−1.3	− 2.2	0.0	0.0	−0.1	− 5.6	+1.2	+ 3.7	− 0.1	− 0.5
2	6.8	24.3	−2.8	− 4.8	+2.1	+18.0	+0.2	+ 9.2	+2.7	+ 8.5	− 0.4	− 1.7
3	10.0	38.4	−5.1	− 9.8	+6.5	+57.7	+4.0	+200.0	−1.5	− 5.0	+11.7	+45.9
4	4.0	16.6	+1.1	+ 2.0	−0.5	− 3.7	+0.1	+ 6.0	+3.0	+ 8.8	− 2.0	− 7.2
5	7.5	27.7	−2.8	− 4.8	+2.1	+18.0	+0.2	+ 9.2	+2.2	+ 6.6	− 0.4	− 1.7
6	6.8	26.7	−3.0	− 5.7	+2.3	+24.3	−0.3	− 9.0	+5.3	+18.7	− 1.0	− 4.2
7	3.3	25.4	+3.5	+ 6.7	0.0	0.0	0.0	0.0	+6.5	+35.3	+11.5	+47.9
8	7.0	25.9	−2.8	− 4.8	+2.1	+18.0	+0.2	+ 9.2	+4.2	+12.8	− 0.4	− 1.7
9	2.3	6.5	+6.0	+10.3	+3.0	+18.2	+1.0	+ 50.0	+1.4	+ 3.4	− 6.8	−23.6

All percentage values are based on results from cylinder diagrams. For example, in Test No. 4 the pressure at cut-off shown by the pipe-card is 2 per cent in excess of that shown by the cylinder or true card; the pressure at release by the pipe-card is 3.7 per cent less than by the true card; the pressure at the beginning of compression by the pipe-card is 6 per cent greater than by the true card; and the mean effective pressure by the pipe-card is 8.8 per cent greater than by the true card.

Comparisons have been made from tests run under still other conditions, and all conclusions thus reached have been consistent with those presented. This whole plan of work was outlined with the

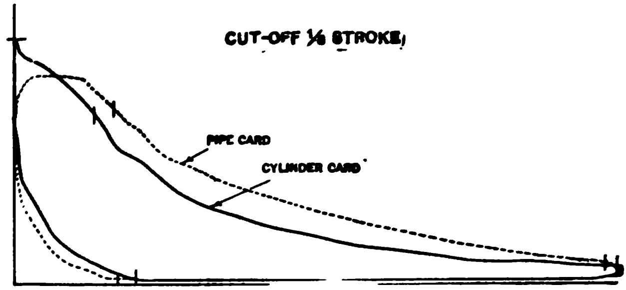


Fig. 169.

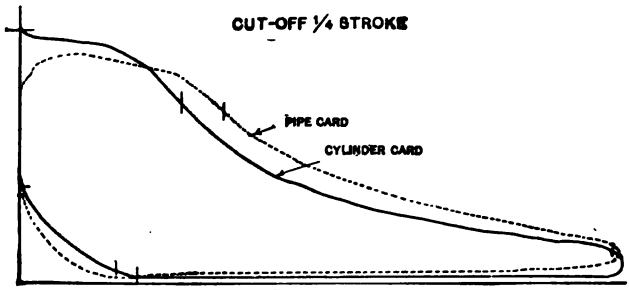


Fig. 170.

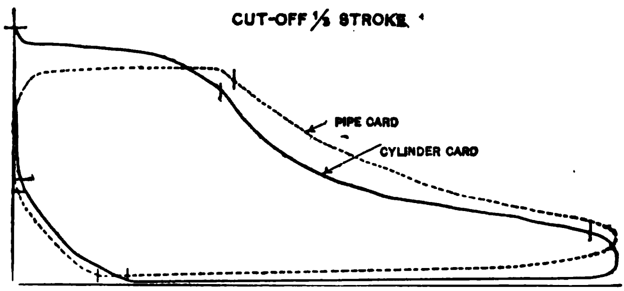


Fig. 171.

Note.—The cut-off as given above is approximate. The steam-pressure (80 pounds), the speed (200 revolutions per minute) and the length of pipe (10 feet), were constant for all diagrams on this page.

expectation of securing such data as would permit a complete analysis of the effects produced by a pipe. But the results show that these effects are modified by so many different conditions that their precise character cannot be safely predicted. Even if it were possible to

construct an expression for reducing a distorted pipe diagram to a form which would correctly represent the relation of pressure and volume within the cylinder, the number of its terms would be so great, and its form so complicated, that the expression would have no practical value.

131. **Conclusions.**—The following conclusions constitute a summary of the data already presented:

1. If an indicator is to be relied upon to give a true record of the varying pressures and volumes within an engine cylinder, its connection therewith must be direct and very short.

2. Any pipe connection between an indicator and an engine cylinder is likely to affect the action of the indicator; under ordinary conditions of speed and pressure a very short length of pipe may produce a measurable effect in the diagram, and a length of three feet or more may be sufficient to render the cards valueless, except for rough or approximate work.

3. In general, the effect of the pipe is to retard the pencil action of the indicator attached to it.

4. Other conditions being equal, the effects produced by a pipe between an indicator and an engine cylinder become more pronounced as the speed of the engine is increased.

5. Modifications in the form of the diagram resulting from the presence of a pipe are proportionally greater for short cut-off cards than for those of longer cut-off, other things being equal.

6. Events of the stroke (cut-off, release, beginning of compression) are recorded by an indicator attached to a pipe later than the actual occurrence of the events in the cylinder.

7. As recorded by an indicator attached to a pipe, pressures during the greater part of expansion are higher, and during compression are lower, than the actual pressures existing in the cylinder.

8. The area of diagrams made by an indicator attached to a pipe may be greater or less than the area of the true card, depending upon the length of the pipe; for lengths such as are ordinarily used the area of the pipe-cards will be greater than that of the true cards.

9. Within limits the indicated power of the engine is increased by increasing the length of the indicator-pipe.

10. Conclusions concerning the character of the expansion or compression curves, or concerning changes in the quality of the mixture in the cylinder during expansion or compression, are unreliable when based upon cards obtained from indicators attached to the cylinder through the medium of a pipe, even though the pipe is short.

CHAPTER XIV.

THE EFFECT OF LEAD UPON LOCOMOTIVE PERFORMANCE.

132. Lead.—When a valve admits steam to the cylinder before the piston has completed its return stroke it is said to have lead. The amount of lead is the width the steam-port is open when the engine-piston is at the beginning of its stroke. Speaking in very general terms it may be said that the effect of lead is to admit steam to the cylinder *before* the piston is ready to start on its forward stroke. Its presence tends to insure an abundance of steam behind the piston, at the very beginning of the stroke, to assist in the maintenance of a satisfactory supply throughout admission and to promote the smooth running of machinery, which is likely otherwise to be noisy. Practice in valve-setting has always recognized the value of lead, and it is not often that an engine is run without it.

Any admission of steam behind the piston before it has completed its return stroke, however, results in negative work, and for this reason the amount of lead should not be excessive. It appears reasonable to assume that the lead should at all times be such as will insure a complete filling of the clearance space, but that more than this will affect unfavorably the economic performance of the engine. But the amount of lead necessary to give this result can only be determined by the use of the indicator, and as the requirement is likely to vary with changes in speed, or with the amount of compression employed, the problem is not a simple one.

In establishing the lead for the valves of a locomotive it is usual to make the measurements when the reverse-lever is in its extreme position This determines the lead for the maximum valve travel. Now, it happens that the most common forms of locomotive link-motion give an increase of lead as the cut-off is shortened. Thus, with a $\frac{1}{16}$-inch lead in full gear a valve may have as much as $\frac{3}{8}$-inch at running cut-off, the exact amount of the increase depending upon

the proportions of the individual gear. For many years it was the practice to make the lead in full gear anywhere from $\frac{1}{16}$ inch to $\frac{1}{8}$ inch, the larger amount being commonly employed upon locomotives in higher speed service. These amounts give satisfactory action when the valve is at full travel, and for a long period the practice was not questioned. At about the time the Purdue testing-plant came into existence, it was suggested that the important thing was to have the lead satisfactory for the running cut-offs rather than at full gear, and that this should be accomplished even at some sacrifice in the distribution for other cut-offs. The advent of the Purdue locomotive gave some opportunity for experiments along this line.

133. Tests Involving Different Amounts of Lead.—Locomotive Schenectady No. 1, as received from its builders, was found to have a $\frac{1}{16}$-inch lead for the full travel of the valve, and at short cut-off to give an indicator-card having a loop at the initial end. As the presence of such a loop is always to be accepted as evidence of an excessive lead or compression, it was determined to change the setting of the valves by changing the position of the "go-ahead" eccentric on the axle by an amount sufficient to eliminate the loop. Before proceeding with this, however, measurements were made to determine all events of the stroke, and four tests were run to determine the performance of the engine. The change was then made, all measurements retaken, and other tests were run. The results of the tests made prior to the change are given in Chapter IV. as Series V, while those obtained after the change constitute Series A. The A setting was, in fact, allowed to stand for all subsequent work with Schenectady No. 1, except in the case of Series J and K. The tests with which the present discussion is immediately concerned are eight in number, designated as follows:

15-1-V	in comparison	with	15-1-A.
25-1-V	"	"	25-1-A.
35-1-V	"	"	35-1-A.
55-1-V	"	"	55-1-A.

134. Effect of Lead upon the Events of the Stroke.—In reducing the lead, the "go-ahead" eccentric was moved backward a distance of about $\frac{1}{4}$ inch, as measured on the circumference of the axle, or through an angle of approximately four degrees. The lead at full gear, which previous to this change had been $\frac{1}{16}$ inch, was in this manner reduced to zero. To secure a measure of the effect of the change

upon the events of the stroke, the engine was run at low speed under a heavy load, with the reverse-lever in each of the several notches from the center to the extreme forward position, for a half-hour, during which time two complete sets of indicator-cards were taken. Cards were taken at low speed and heavy power, defining clearly the events of the stroke. All cards obtained by the process described were worked up to show the per cent of stroke at which the events occurred, and the average values for all cards taken with the reverse-lever in a given position were accepted as representing the events, for that position. These values are given in Table LX., in which Series V represents conditions before the change, and Series A the conditions after the change. They are also shown graphically by Fig. 172.

TABLE LX.

EVENTS OF STROKE IN PER CENT MEASURED FROM INDICATOR-CARDS.

Notches Forward of Center.	Series V.				Series A.			
	Admission.	Cut-off.	Release.	Compression.	Admission.	Cut-off.	Release.	Compression.
1	5.04	25.1	67.6	42.1	3.25	24.7	71.1	33.1
2	3.65	33.3	73.8	31.7	1.7	33.7	77.9	28.6
3	2.6	42.7	78.6	25.4	1.16	45	81.6	23.5
4	1.74	51.4	82.5	20.8	.69	54.1	85.7	17.7
5	1.16	59.3	85.8	17	.41	61.9	89	14.6
6	.74	66	88.5	13.7	.16	69.8	91	11.2
7	.5	71.6	90.7	10.8	.1	74.8	93.1	9.39
8	.21	76.1	92.6	9.06	.14	79.3	94.2	7.5
9	.16	79.8	93.7	7.45	0	82.9	95.4	6.04
10	.05	83	94.9	6.12	0	84.1	96.2	5.5
11	0	85.1	95.8	5.00	0	86.71	97	4.4
12	0	87.3	96.6	4.19	0	88.56	97.5	3.87
13	0	88.7	97.2	3.52	0	90.2	98.0	2.77
14	0	90	97.7	3.02	0	91.3	98.6	2.19
15	0	91.1	98.1	2.76	0	92.7	98.9	2.09

It will be seen that the effect of the change in valve-setting is to retard all the events of the stroke. For long cut-offs these effects are so small as to be negligible, but as the cut-off is shortened they increase in all cases except with reference to the cut-off. From a purely academic point of view these effects appear to be advantageous. For example, with a given cut-off the release is delayed, thus prolonging the expansion, while a relatively greater delay in the beginning of compression contributes to a free exhaust. With reference to admis-

sion, it appears that when the reverse-lever is well forward (tenth to fifteenth notch) neither the original lead of $\frac{1}{16}$ inch nor the absence of it have a sufficient effect on the admission to show on the indicator. The reason for this is probably to be found in the fact that at long cut-offs the travel of the valve is so rapid, when the piston is at the end of its stroke, that the precise time when the port opens is not a matter of great importance. As the reverse-lever is moved toward the center,

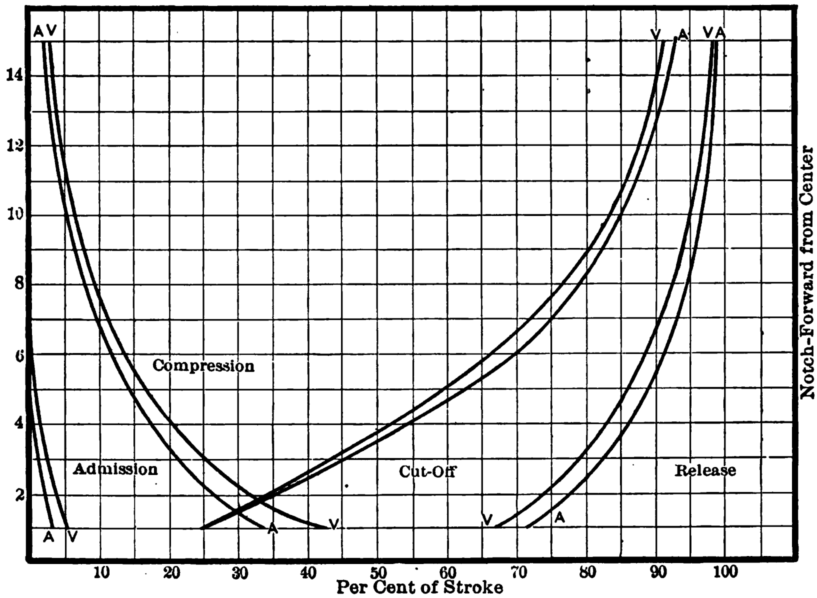


FIG. 172.—Events of Stroke.

however, the lead increases, and, as will be hereafter shown, the maximum port-opening diminishes, and the effect of the lead upon admission, both relatively and actually, is increased.

135. Effect of Lead upon Valve-travel and Port-opening.—The extent of the valve travel for the several positions of the reverse-lever was determined by means of apparatus shown in Fig. 173. This consisted of a stop (*a*) secured to the valve-stem, a wooden straight-edge (*b*) secured to the engine frame directly above and parallel to the valve-stem, and a try-square (*c*) with a blade sufficiently long to have contact with the stop as it moved back and forth with the valve-stem. With the locomotive running under desired conditions the try-square was moved on the straight-edge till the

stop just touched it at one end of its stroke, after which a line showing its location was drawn on a piece of paper fastened to the straightedge. The try-square was then moved to the other end of the straightedge and the process repeated, after which the actual travel of the

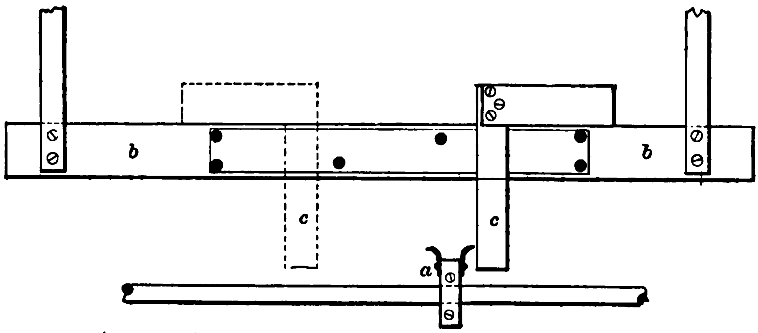


FIG. 173.

valve was found by measuring the distance between the two lines and subtracting the known width of the stop.

The average port-opening was found by subtracting twice the outside lap of the valve from the travel of the valve and dividing by two. Table LXI. presents the record of valve travel and the port-openings, as determined in this way for each position of the reverse-lever.

TABLE LXI.

VALVE TRAVEL AND PORT-OPENING.

(From Right Cylinder of Locomotive.)

Notch Forward of Center.	Valve Travel, Inches.		Port-opening, Inches.	
	Series V.	Series A.	Series V.	Series A.
1	1.94	1.88	.22	.19
2	2.04	1.96	.27	.23
3	2.19	2.13	.34	.31
4	2.37	2.32	.43	.41
5	2.58	2.54	.54	.32
6	2.72	2.81	.61	.65
7	3.07	3.06	.78	.78
8	3.35	3.30	.93	.90
9	3.55	3.62	1.02	1.06
10	3.94	3.90	1.22	1.20
11	4.28	4.28	1.39	1.39
12	4.59	4.58	1.54	1.54
13	4.90	4.91	1.70	1.70
14	5.22	5.25	1.86	1.87
15	5.57	5.57	2.03	2.03

An examination of the table discloses the fact that at short cut-offs, when the lead is reduced, the valve travel and, consequently, the port-opening are also reduced, though the effect in these particulars is not great. The table is perhaps chiefly of interest for the actual values it assigns to the port-openings for the different positions of the reverse-lever. A moment's consideration will show that the lead at short cut-offs does not need to be great to equal the maximum port-opening, in which case the maximum opening of the port occurs before the piston starts on its stroke. With lead at short cut-offs, of a quarter of an inch or more, it is evident that such a condition does occur. The precise manner in which it occurs is best seen from the ellipse (Fig. 191), showing the relative motion of the valve and piston.

136. Effect of Lead upon the Form of Indicator-cards.—Representative cards from the several tests are reproduced as Fig. 174, and have already been referred to. As previously stated, it was the purpose in changing the valve-setting to so reduce the lead that the loop which appears in the cards of Series V should disappear. The results show (cards of Series A) that this was not entirely accomplished, and that some further reduction might have been made upon the right-hand side, but the change which was made is nevertheless significant in its effect upon the indicator-cards.

First to be noticed is the smaller size of the cards of Series A. It is clear that by delaying the opening of the port the amount of steam admitted is reduced and the work per stroke diminished. It must be admitted, also, that the cards of Series A are not so well filled out at the initial end as are those of Series V, and that they are not so well marked by the events of the stroke. These effects are shown by Fig. 175, in which the dotted outline represents a card taken when the lead was considerable, and the full line a card taken after it had been reduced. The reduction of lead effects a reduction in the back-pressure losses, but it also reduces the positive work under the steam and expansion lines. Judging from the cards alone it would appear to be a fair question as to whether any advantage had been gained by reducing the lead; but it must be remembered that a reduction in the size of a card does not necessarily constitute an argument against reduced lead, since the work per stroke can always be increased by moving forward the reverse-lever. The advisability of the change is rather to be judged by the amount of steam consumed per unit power, and to this phase of the matter attention will now be given.

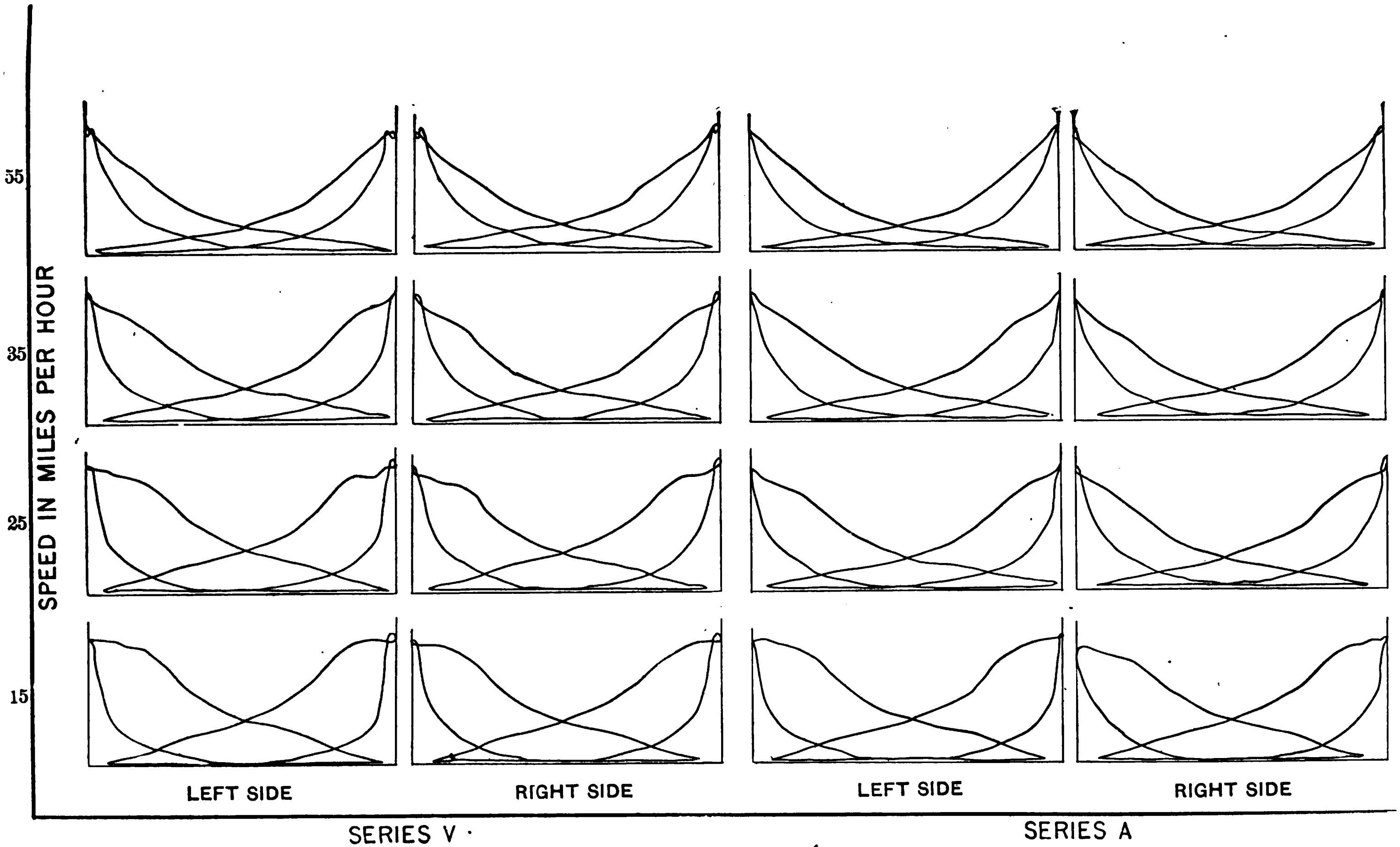

SPEED IN MILES PER HOUR
55
35
25
15
LEFT SIDE
RIGHT SIDE
LEFT SIDE
RIGHT SIDE
SERIES V
SERIES A

FIG. 174.

137. Steam Consumption.—The steam consumption constitutes the one fact which finally determines the usefulness of any alteration in steam distribution. The results of tests, as set forth in Table

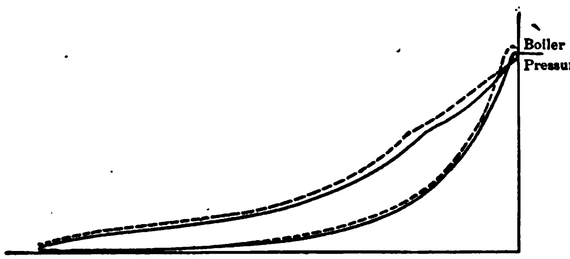


FIG. 175.

LXII. and as presented graphically by Fig. 176, show conclusively that the reduction in lead effected a reduction in the amount of steam consumed. It is often assumed that high speeds admit of the use of

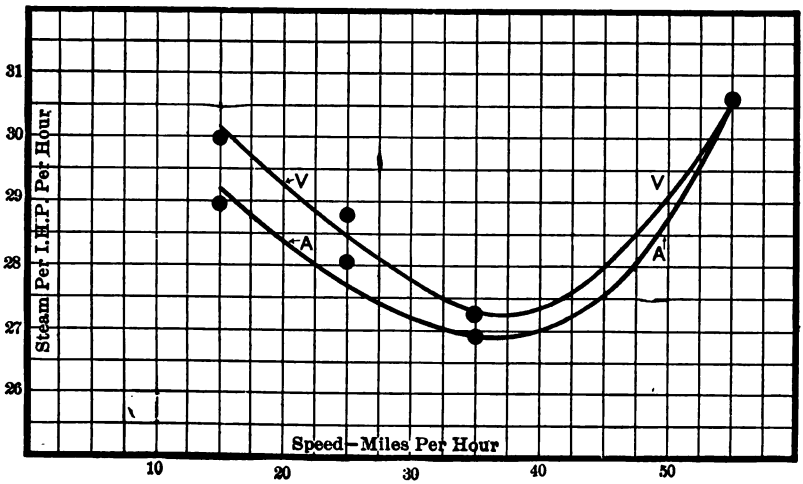


FIG. 176.—Steam Consumption.

more lead than is desirable when the speed is low, but the results do not confirm such an assumption. They show that the loss arising from excessive lead is less than when the speed is lower, but they do not in any case show that the larger lead gives the greater economy.

From the standpoint of steam consumption per indicated horsepower, there can be no question but that the performance of the engine was improved by so reducing the lead as to entirely remove

the loop from the initial end of the card. The change appears to have been justifiable, even though it resulted in making the initial end of the indicator-card somewhat rounded under certain conditions of running (Fig. 174). A comparative study of the indicator-cards and the record of steam consumption will be found profitable.

TABLE LXII.

STEAM CONSUMPTION.

Speed, Miles per Hour.	Approximate Cut-off, Per Cent.	Steam per I.H.P.	
		Series V.	Series A
15	..	29.97	28.92
25	..	28.78	28.06
35	25	27.27	26.93
55	..	30.62	30.64

138. Lead and Machine Friction.—It is shown elsewhere (Chapter XIX.) that high initial pressure is attended by heavy losses between the cylinder and the draw-bar in the form of machine friction, from which it is fair to conclude that the friction was greater in the tests of Series V than in those of Series A. If this were actually the case, and if the steam consumption per draw-bar horse-power were plotted, the difference between the two series would be greater than that obtained by basing the comparison on the indicated horse-power, that is, greater than represented by Fig. 176.

139. Conclusions.—Although the experiments above discussed are somewhat limited in scope, the results justify the following rather general conclusions:

1. Lead should never be so great as to give a loop at the initial end of the indicator-card.

2. If necessary to avoid the loop at running cut-offs, the valves of a locomotive should be set with no lead, and even with negative lead for the full-stroke position of the reverse-lever.

3. Locomotives having their valves set with no lead in full gear will be found to respond to the throttle promptly under starting conditions. The valves move so rapidly, when the crank is passing its center, that the port opening will be liberal before the piston is fairly started on its stroke.

CHAPTER XV.

THE EFFECT OF OUTSIDE LAP UPON LOCOMOTIVE PERFORMANCE.

140. **Outside Lap.**—The term "outside lap," or "steam lap," refers to the amount by which a slide-valve in its central position overlaps the outside edges of the steam-ports. Thus, in Fig. 177, the outside lap of the upper valve is ¾ inch, of the middle valve, 1 inch, and of the lower valve, 1½ inch. Perhaps the most obvious effect of increasing the outside lap is the necessity for increased valve travel, if the cut-off is to remain unchanged. A casual inspection of Fig. 177 will show that a certain movement of the valve with ¾-inch outside lap may be sufficient to operate the engine, and yet, when applied to the valve having 1½-inch outside lap, prove inadequate to open the steam-ports.

To determine the effect upon the power and efficiency of a locomotive of different amounts of outside lap, three series of tests were run, designated as A, K, and J respectively. The slide-valves employed in each series were alike in every respect, except as to the outside lap, which varied as follows:

Series A, Outside Lap	¾ inch
" K, " "	1 "
" J, " "	1½ "

Fig. 177 shows all the dimensions.

No change was made in the setting of the eccentric or in any other position of the valve-gear, when one valve was interchanged for another having a different amount of outside lap. When increased travel was necessary, it was obtained by carrying the reverse-lever in a notch further from the center of the quadrant. It was sought to have tests run under conditions which would give as nearly as possible identical cut-offs. This result was not perfectly secured, but the degree of success is shown by Table LXIII., giving the events of the stroke as obtained from the indicator-cards.

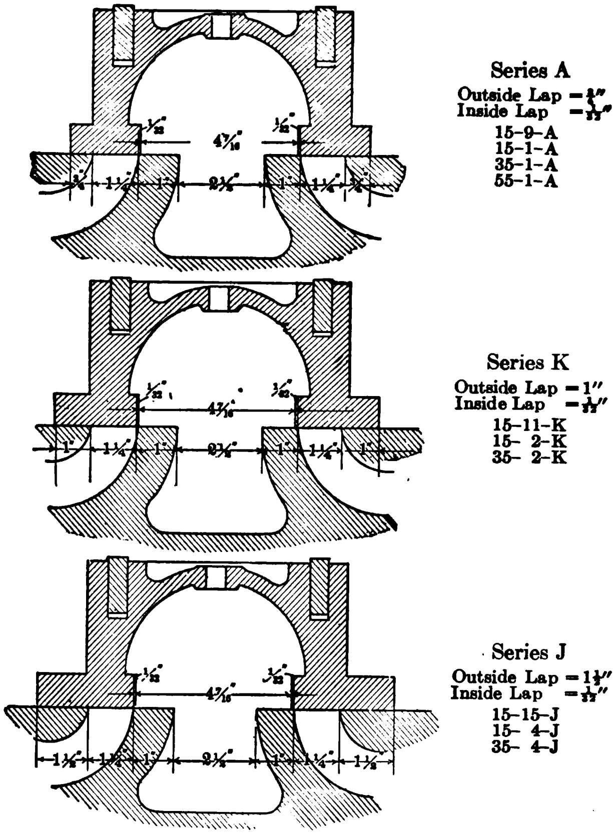


Fig. 177.

TABLE LXIII.

EVENTS OF STROKE.

Approximate Cut-off.	Outside Lap, Inches.	Reverse-lever, Notches Forward of Center.	Actual Events of Stroke.			
			Cut-off, Per Cent.	Release, Per Cent.	Compression, Per Cent.	Admission, Per Cent.
One Quarter	¾	1	23.4	65.9	36.0	4.3
	1	2	26.3	69.7	32.8	2.8
	1½	4	23.3	69.9	31.6	1.3
Three Quarters	¾	9	82.5	95.1	5.7	0
	1	11	79.5	93.5	7.1	0
	1½	15	69.4	90.5	10.1	0

141. Events of Stroke.—In order to study the events of the stroke in more detail there was attached to the engine a valve-motion indicator, by means of which the diagrams were made, showing the relative motion of the valve and piston. An analysis of one of these valve-motion diagrams * (Fig. 178) which as presented are approximately one fourth their full size, will disclose effects resulting from increased outside lap as follows: The rapidity with which the valve opens the steam-port is increased, resulting in a freer admission. The range of cut-off is decreased. When the cut-off is short the exhaust is hastened, an effect which diminishes as the cut-off is lengthened, and may disappear entirely when the reverse-lever is in an extreme position. The amount by which the steam-port is opened to the exhaust is, with short cut-off, increased, the extent of such increase being approximately equal to the increase in the outside lap. This applies, however, only to the shorter cut-offs and to reasonably small outside laps, since, when either of these factors becomes large, the exhaust-port of the valve overtravels the steam-port, and exposes thereby its full width beyond which it is not safe to go.

142. Changes in the Form of Indicator-cards Resulting from Changes in Outside Lap.—The cards for the several tests, as actually obtained, are shown by Figs. 180, 181, and 182. Referring to those taken at a speed of 15 miles an hour (Fig. 180), it will be seen that, as regards cut-off, they are nearly identical, but differ essentially in other respects. The cards made for the valves of greatest lap have a higher steam line and lower exhaust and compression lines than those taken when the laps was less. The result is a card of greater breadth at the initial end, and of greater area for the same cut-off. The same characteristics are to be seen in the cards taken at 35 miles per hour (Fig. 181), though in these cards it is noticeable that the cut-offs are not quite the same, that for which the valve had 1 inch outside lap being greater than for the other two.

Cards representing long cut-off (Fig. 182) are hardly to be regarded as comparable, since, owing to the fact that the reverse-lever was in its extreme forward position, the cut-off was regularly diminished as the lap was increased. The effect noted, therefore, with increase in outside lap, is influenced by the corresponding decrease in cut-off. Cards are nevertheless given, since they emphasize the fact that even with the outside lap as great as 1½ inches, the admission is prompt, and the steam line well sustained, when the reverse-lever is nearly in

* For a detailed description of a valve-motion diagram, see paragraph 154.

FIG. 178.—Valve Diagrams for Notches 1, 2, and 4.

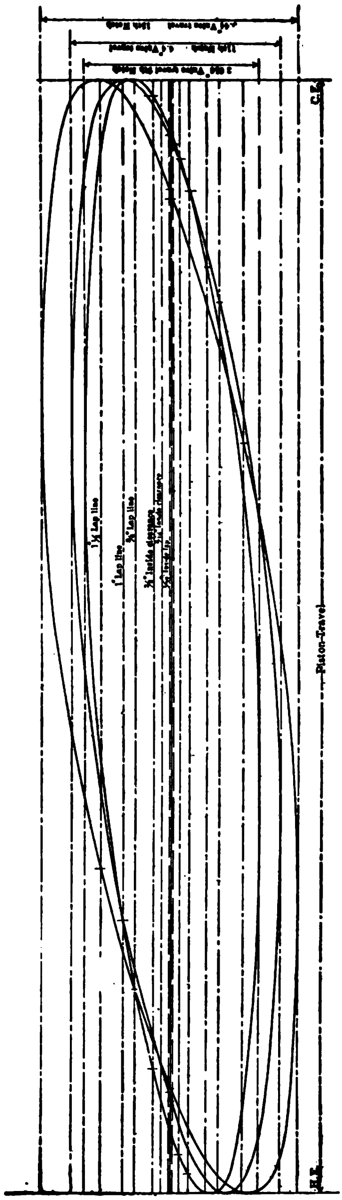


FIG. 179.—Valve Diagrams for Notches 9, 11, and 15.

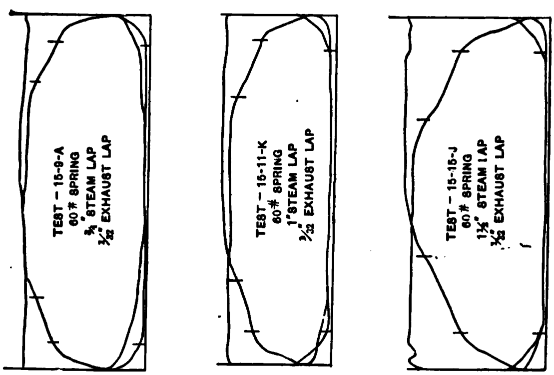


FIG. 180.—Average Cards 15 M.P.H. Early Cut-off.

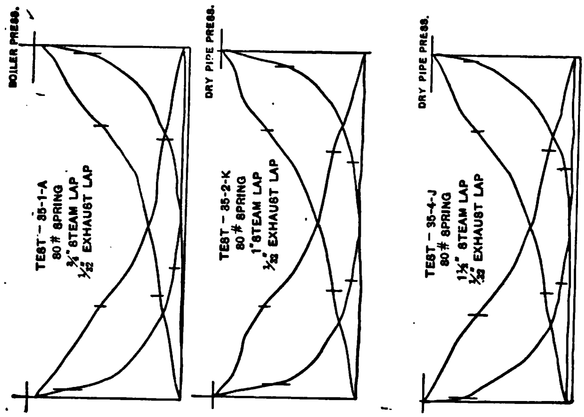


FIG. 181.—Average Cards 03 M.P.H.

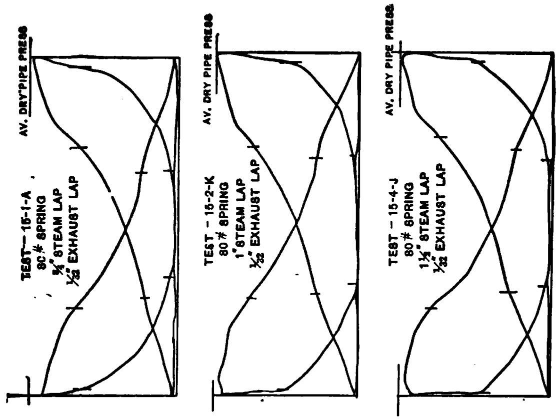


FIG. 182.—Average Cards 15 M.P.H. Late Cut-off.

its extreme position. The lower cards of this set (Fig. 182) show the longest cut-off which can be secured with a valve of the proportions given. They well illustrate the difficulty of getting, in connection with large outside lap, a card which shall be full at both ends, a very desirable thing in locomotive service.

A comparison of the cards of Figs. 180, 181, and 182 discloses the general effects of increasing the outside lap, or these may be more easily judged by reference to Fig. 183, which is to be regarded as a diagrammatic representation of what actually takes place. In this figure the full line is the normal card, such as may result from $\frac{3}{4}$-inch outside lap, and the dotted line is that which results when a valve is used having a greater amount of outside lap. When the cut-off remains constant, the effect upon the form of the card of increasing

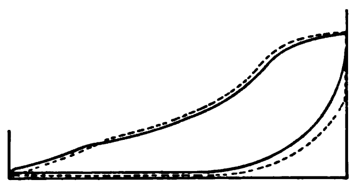


FIG. 183.—Effect of Outside Lap on the Form of an Indicator-card.

the outside lap is to raise the steam and expansion lines and to lower the exhaust and compression lines. It gives a freer exhaust, but it reduces the possible range of cut-off. All of these effects may be traced in the actual cards (Figs. 180, 181, and 182), and are what we should expect from a consideration of the valve diagrams.

143. Power Variation.—The changes in power, resulting from changes in outside lap, are not of great importance, since, within limits, the power is at all times under the control of the engineer who has but to manipulate the reverse-lever in such a manner as to serve his purpose. For this reason, and because the similar cards are not identical in cut-off, no attempt should be made to compare numerical values representing M.E.P. or power.

144. Steam Consumption.—Evidently the important fact is that of steam consumption. If the engine is more efficient under one condition of lap than under another, the argument will be strong in favor of adopting that condition. The steam consumption, as shown by the several tests, is shown by Table LXIV.

TABLE LXIV.

STEAM CONSUMPTION.

Speed, Miles per Hour.	Approximate Cut-off, Per Cent of Stroke.	Actual Cut-off, Per Cent of Stroke	Outside Lap, Inches.	Steam per I.H.P. per Hour.
15	80	82.5 79.5 69.4	¾ 1 1½	39.2 37.8 35.1
15	25	23.4 26.3 23.3	¾ 1 1½	28.9 28.8 28.5
35	25	23.4 26.3 23.3	¾ 1 1½	26.9 27.8 26.8

The table shows a gain in steam consumption with increased outside lap while running at slow speed and long cut-off, but in drawing any conclusions it must be borne in mind that as the lap was increased the cut-off was materially decreased, a fact in itself sufficient to account for the better economy. At running cut-offs and at ordinary speeds there seems to be very little change in the steam consumption, whatever the lap.

The fact that any increase of outside lap does not increase the economy of the engine, while it does materially reduce the flexibility of the valve-gear in respect to range of cut-off, is sufficient to justify common practice which makes this feature of the locomotive slide-valve comparatively small.

CHAPTER XVI.

THE EFFECT UPON LOCOMOTIVE PERFORMANCE OF INSIDE CLEARANCE.*

145. **Inside Clearance.**—A slide-valve, which in middle position has its inside edges in line with the inside edges of the steam-ports, over which it is designed to travel, has neither inside lap nor inside clearance. It is sometimes referred to as being "line and line." When the valve is so proportioned as to allow the inside edges of the valve to overlap the inside edges of the steam-ports for mid-position, then the valve has inside lap; while, on the other hand, if the inside edges of the valve fail to cover the steam-ports, the valve has inside clearance. Thus, in Fig. 184, the first valve shown has $\frac{1}{32}$-inch inside lap; the second, $\frac{3}{16}$-inch inside clearance; and the third, $\frac{3}{8}$-inch inside clearance.

The effect of changing a valve from inside lap to inside clearance, other things remaining unchanged, is to hasten release and to delay compression, and hence to increase the interval of time during which the exhaust-port is open. It also increases the extent of exhaust-port opening. As a consequence of these effects, the exhaust is made freer and back pressure is reduced, giving an advantage in the operation of the engine, which is greatly desired, and which would be accepted

* Acknowledgment for assistance rendered in the investigations of inside clearance and outside lap is due to Messrs. Halstead, Robinson, Ede, and Wilson, who, while students, presented the facts which are herein summarized, in the form of the following theses:

"The Effect of Increased Inside Clearance upon the Efficiency of Locomotive 'Schenectady,'" by W. G. Halstead, B.M.E., 1897; "The Effect of Increased Outside Lap upon the Efficiency of Locomotive 'Schenectady,'" by G. P. Robinson, B.M.E., 1897; "The Effect of Changes in Inside Clearance upon the Efficiency of Locomotive 'Schenectady No. 1,'" by S. S. M. Ede, B.M.E., 1903; "The Effect of Changes in Valve Proportion upon the Efficiency and Power of Locomotive 'Schenectady No. 1,'" by A. M. Wilson, M.E., 1903.

without question if it were not for the assumption that loss of efficiency attends the earlier exhaust. Until quite recently it was the practice to run all locomotives with a small amount of inside lap, usually $\frac{1}{16}$ of an inch. With the advent of the modern engine, how-

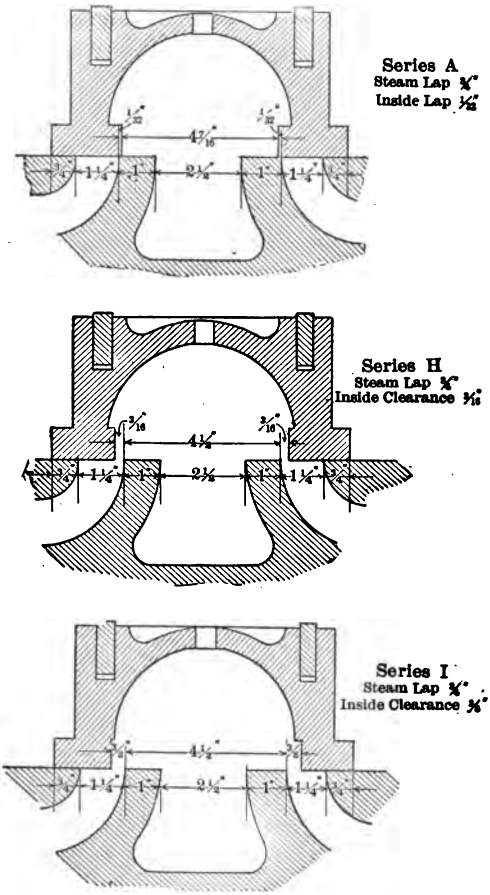


FIG. 184.

ever, and especially in response to a demand for a free-running engine at higher speeds, the inside lap has been reduced, and in some cases an $\frac{1}{8}$-inch or $\frac{3}{16}$-inch inside clearance has been employed. But as to the wisdom of such practice men best qualified to speak have differences in their opinions.

For the study of this subject three valves having the proportion

shown by the three views of Fig. 184 were employed. These valves are identical, excepting as to their inside dimensions. Six tests were run with each set of valves, the conditions of speed and cut-off being those which are set forth by Fig. 185. All valves were tested with the same setting, the position of eccentrics and reverse-lever being identical for the corresponding tests of the different series. The tests of each series, therefore, were made duplicates of each other, except in so far as the results were effected by the changed proportion of the valves. We may now consider the nature and extent of these effects.

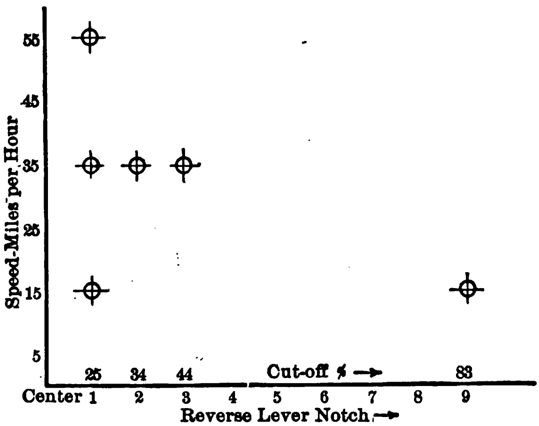


FIG. 185.—Tests Run.

146. Maximum Opening of Steam-port into Exhaust. — For short cut-offs the travel of the valve is ordinarily insufficient to open the full width of the steam-port into the exhaust. In such cases cutting the valve away to give inside clearance increases the maximum opening into the exhaust by an amount equal to the change. A very material gain in exhaust-opening may thus be secured. For example, the tests which were run with the reverse-lever in the first notch gave a cut-off of 25 per cent. The maximum opening of the steam-port into exhaust under these conditions for the several valves tested were as follows:

For valve having $\frac{1}{32}''$ inside lap (Series A)	$\frac{5}{8}'' = \frac{20}{32}''$
" " " $\frac{3}{16}''$ " clearance (Series H)........	$\frac{3}{4}'' = \frac{24}{32}''$
" " " $\frac{3}{8}''$ " " (Series I)........	$1\frac{1}{32}'' = \frac{33}{32}''$

147.—Changes in the Events of the Stroke Resulting from Inside Clearance.— Release and beginning of compression only are affected. The changes in these, with changes in valve clearance, are most pronounced for the shorter cut-offs. Thus, with the reverse-lever in the first notch forward of the center (cut-off, 25 per cent), the relation between clearance, cut-off, and release is shown by Table LXV.

TABLE LXV.

EVENTS OF THE STROKE, CUT-OFF, 25 PER CENT.

Inside Lap or Clearance, Inches.	Release, Per Cent of Stroke.	Beginning of Compression, Per Cent of Stroke
$\frac{1}{32}$ lap	71.1	33.3
$\frac{3}{16}$ clearance	59.0	23.9
$\frac{3}{8}$ clearance	49.4	17.3

The table shows that a change from $\frac{1}{32}''$ inside lap to $\frac{3}{16}''$ inside clearance, a total change of $\frac{1}{8}$, hastens the release and delays the beginning of compression by approximately 10 per cent; this for short cut-off. When the cut-off is lengthened, and the travel of the valve is thereby increased, the relative effect upon the events of the stroke of slight changes in the dimensions of the valve itself becomes less pronounced.

148. Changes in the Form of the Indicator-card Resulting from Inside Clearance.—Fig. 186 presents cards obtained with the reverse-lever in the first notch (25 per cent cut-off), and at speeds of 15, 35 and 55 miles an hour. The full-lined cards (Series A) were obtained in connection with valves having $\frac{1}{32}''$ inside lap, and are assumed as a basis of comparison. The dotted-lined cards on the left (Series H) were obtained with $\frac{3}{16}''$ inside clearance, and on the right (Series I), with $\frac{3}{8}''$ inside clearance. Where slight variations in the steam pressure for the different tests have appeared, the cards of Series H and I have been reduced to equivalent cards, for which the pressure at cut-off is identical with that of the corresponding card of Series A, thus supplying a logical basis for comparison. Referring to Fig. 186 it appears, first, that with increased inside clearance, the earlier exhaust effects a noticeable reduction in the area of the card under the exhaust line; secondly, that the wider exhaust-port opening and the later exhaust closure are responsible for a significant change in the location and form of the compression line, effecting a material increase in the area of the card over the same; and, third, that the reduced compression tends to diminish the maximum height of the

card, so that, as valve clearance is increased, the maximum pressure is reduced. With a given amount of valve clearance all of these effects become more pronounced as the speed is increased. The per cent of area lost under the exhaust line and gained over the com-

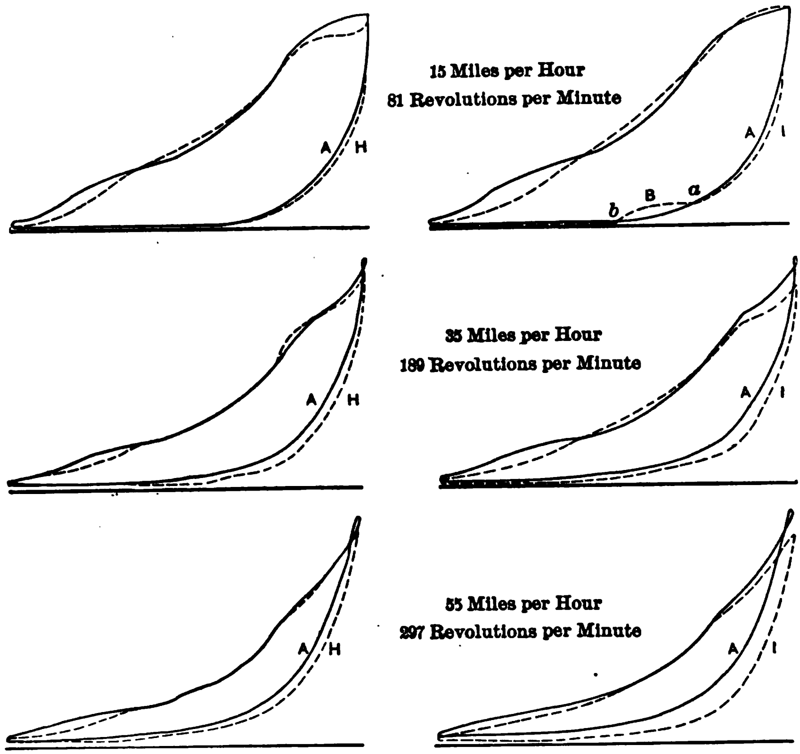


FIG. 186.—Effect of Inside Clearance on the Form of Indicator-cards.

pression line respectively, expressed in terms of the area of the full-lined card, is as given in Table LXVI. The values of the Table appear to justify the conclusion that, when the reverse-lever is in running position, the loss in the area of the card, even at slow speeds, is so small as to be negligible, whereas for all but the slowest speeds there is gain, and at high speed the gain is material.

The changes in the form of the cards under starting conditions are shown by Fig. 187, in which, as before, the full line represents the card taken in connection with the valve having $\frac{1}{32}''$ inside lap (Series A), and the dotted lines cards representing $\frac{3}{16}''$ and $\frac{3}{8}''$ inside clearance

(Series H and I respectively). These cards represent the tests at the ninth notch (cut-off, 80 per cent) and a speed of 15 miles an hour. They show that the loss of area at the ends of the cards, attending increase of clearance, while actually large, is in part compensated

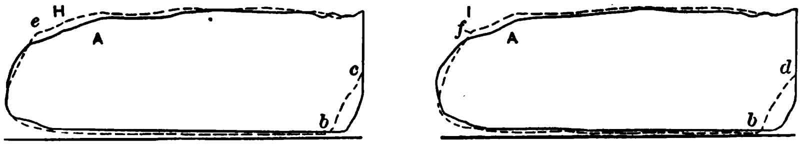


FIG. 187.—Effect of Inside Clearance Under Starting Conditions.

for by the lower exhaust line and, compared with the area of the card, is after all small.

TABLE LXVI.

REVERSE-LEVER FIRST NOTCH FORWARD OF CENTER, CUT-OFF, 25% OF STROKE.

Speed.		Clearance, Inches.	Area Lost by Early Release in Per Cent.	Area Gained by Reduced Back Pressure and Late Compression in Per Cent.	Net Gain (+) or Loss (−) in Per Cent.
Miles.	Revolutions per Minute.				
15	81	$\frac{3}{16}$	3.5	2.5	−1
		$\frac{3}{8}$	6.0	.5	−5.5
35	188	$\frac{3}{16}$	3.5	12.75	+9.25
		$\frac{3}{8}$	5.25	15.0	+9.75
55	296	$\frac{3}{16}$	5.25	21.0	+15.75
		$\frac{3}{8}$	8.75	48.25	+39.25

149. The Blowing through Effect.—The fact that a valve having inside clearance opens both steam-ports to each other as well as to the exhaust-ports (see Fig. 184) has often been regarded as the chief objection to its use. The interval of time during which the two ports are in communication depends not only upon the extent of the clearance itself, but upon the travel of the valve and the speed of the engine. An increase either in the travel of the valve or in the speed of the engine diminishes the interval.

It is important to note also that the intercommunication of the steam-ports does not lead to losses of live steam. Throughout the interval during which the two steam-ports are in communication

both are seeking to exhaust steam. The only effect, therefore, which can result from the so-called blow-over action is an interference between the exhaust from one end of the cylinder and that from the other end; a portion of the steam from the port which is just opening, having a higher pressure than that which is issuing from the opposite port which has been longer open, follows the exhaust cavity of the valve and enters the opposite port, from which steam would otherwise be flowing (Fig. 188). With this understanding, and with the expectation of seeing the interference manifest itself by a rise in the exhaust line, we may now examine the card for the purpose of ascertaining to what extent such action really occurs.

Most important, of course, is an understanding of the action under

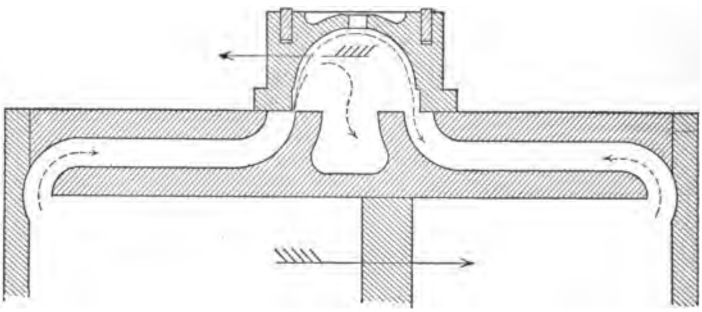


FIG. 188.

running conditions, the cards for which appear as Fig. 186. Here, it appears, that at a speed of 15 miles an hour, $\frac{3}{16}''$ clearance produces no noticeable disturbance of the exhaust line, but when the clearance is increased to $\frac{3}{8}''$, an amount in excess of any which has thus far been commonly employed in locomotive service, a decided hump appears. Thus, referring to the dotted outline representing this condition, the exhaust-port did not close until the point *a* was reached, the sudden rise in back pressure in the vicinity of *b* being due to an influx of steam from the opposite port which was then exhausting. This, then, is the blowing-through effect. Upon the cards representing the 35-mile tests, the effect gives to the exhaust line only a wave-like appearance; and when the speed has been increased to 55 miles, even this, as a distinct characteristic in the form of the card, nearly or quite disappears. No doubt, however, the action does take place even at the higher speed, and its complete suppression would further lower the later part of the exhaust

line and the compression line, but any harmful effect upon the form of the card is entirely lost in the lower sweep of the line which marks the freedom of the exhaust action as a whole.

Those who have feared serious loss in the effect of inside clearance under conditions of starting will be reassured by reference to Fig. 187. As would be expected, the effect here is most pronounced, but with the long cut-offs it occurs so near the end of the stroke that it does not greatly reduce the area of the card. Thus, in Fig. 187, in the dotted diagrams, the rise beginning at *b*, near the end of the exhaust line, appears to be compression. That it is not the result of compression must be apparent when it is remembered that increased clearance delays compression, and hence the exhaust closure for the cards, which are shown dotted, must be later than for the cards which are shown full. As a matter of fact the exhaust-ports for the dotted cards closed at the points *c* and *d* respectively, and consequently most of the rise which occurs at an earlier point in the stroke is the result of steam coming from the opposite steam-port. Obviously the exhaust from the port at the other end began when the return stroke for the end under consideration had reached the points *e* and *f* respectively.

The conclusion is, therefore, that the fact that inside clearance places the two ports in communication is not significant so far as its effect upon the form of the card is concerned, even though the amount of clearance be as great as $\frac{3}{8}''$. When the cut-off is late the effect is too near the end of the stroke to be serious, and when the cut-off is early, it is of moment only at slow speed, and even then the increased area of port-opening which the clearance gives serves to quickly dissipate the back pressure resulting from the blow-over.

150. Mean Effective Pressure.—It has already been shown that increasing the inside clearance will, at speed, increase the area of the card, or, in other words, the mean effective pressure and, consequently, the power of the engine. A measure of such increase for speeds of 55 miles an hour with the reverse-lever in the first notch (cut-off, 25 per cent) is as follows:

	M.E.P.
With valve having $\frac{1}{32}''$ inside lap (Test 55–1–A).............	17.56
With valve having $\frac{3}{16}''$ inside clearance (Test 55–1–H)........	21.58
With valve having $\frac{3}{8}''$ inside clearance (Test 55–1–I).........	23.68

An increase in mean effective pressure, and, consequently, of the power of the engine in the ratio of 17.5 to 23.7, or more than 30 per

cent, by merely cutting out the inside of the valve, does, in fact, constitute a strong argument in favor of a more general clearance. In urging the force of such an argument, however, it should, of course, be granted that such increase is only possible at high speed; but it is when speeds are high that those effects which are obtainable by means of increased clearance are most desired.

While the conclusions of the preceding paragraphs are important and true, too much emphasis should not be given card areas. The fact that clearance helps to make a larger card is but a part of the story, for the effects of a large card may be obtained in various other ways, as, for example, by a change in the position of the reverse-lever. For this reason it is unfair to assume that any device or expedient which increases the card is justified.

With this in mind may be noted the character of the changes in the form of the card resulting from clearance. The maximum pressure, the back pressure, and the pressure during compression are all thereby reduced. Pressures acting upon the piston are practically nowhere increased, but, on the other hand, for the most part reduced. In other words, the increase in effective work is chiefly the result of a diminution in the negative work of the cycle. Herein is the explanation of the fact, which all experimenters with inside clearance have observed, namely, that "it makes a free-running engine."

But the test by which any apparatus, or system of adjustment, which is designed to improve the distribution of steam within an engine cylinder should be judged, is that of economy. The real question is whether by its use a given amount of power can be obtained by the expenditure of a smaller amount of steam. If so the device will prove economical, and will also raise the maximum limit of power which the locomotive can deliver. If it cannot it will prove wasteful, and will reduce the maximum power which the locomotive can deliver. The effect, therefore, of inside clearance on steam consumption is a matter which should outweigh other considerations.

151. Steam Consumption as Affected by Increased Clearance.—A detailed statement of the steam consumption, and of other related facts under the conditions of the several experiments, is set forth in full in Table LXVII., while a summary of the important facts is presented as Figs. 189 and 190. Fig. 189 shows the steam consumption of the engine at different rates of speed, and proves the truth of the commonly accepted theory, namely, that increased inside clearance results in loss of efficiency at low speeds. For example, changing the

inside of a valve from $\frac{1}{32}''$ lap to $\frac{3}{16}''$ clearance increases the steam consumption at 15 miles an hour one pound per horse-power, and a further increase of inside clearance to $\frac{3}{8}''$ adds three pounds to the steam consumption. As the speed is increased, however, the differ-

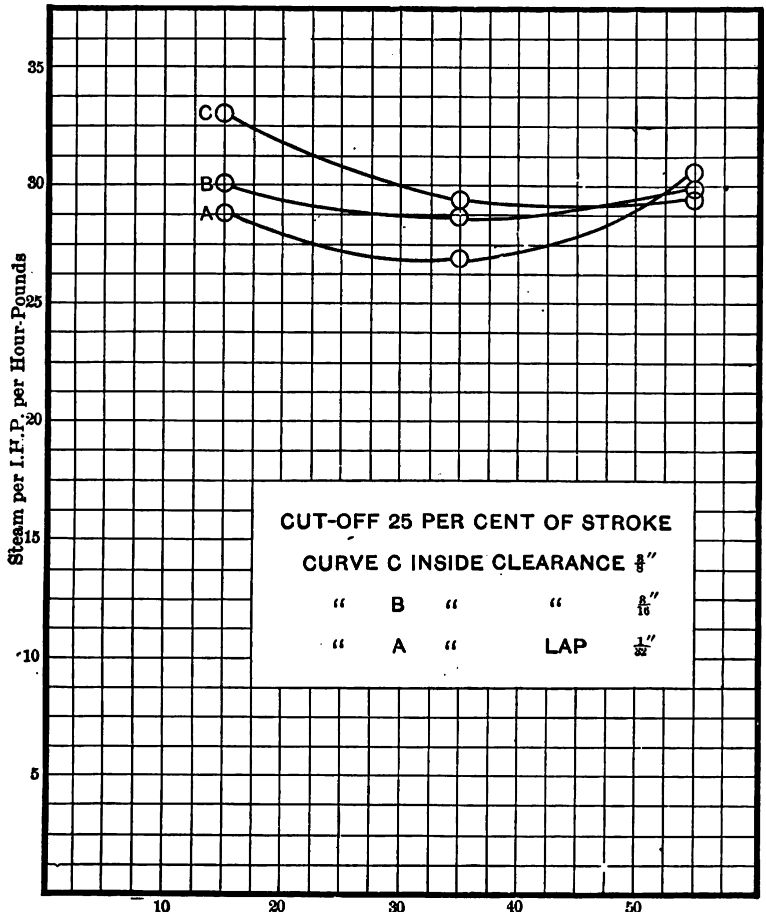


Fig. 189.—Inside Clearance and Steam Consumption.

ence in steam consumption for the different amounts of inside clearance diminishes, until at a speed of 50 miles an hour (270 R.P.M.), or thereabouts, we have the same steam consumption for all cases. For speeds above 50 miles an hour the least steam consumption attends the use of the greatest amount of inside clearance, while the steam

consumption for the valve having no inside clearance increases rapidly, with increase of speed beyond this limit.

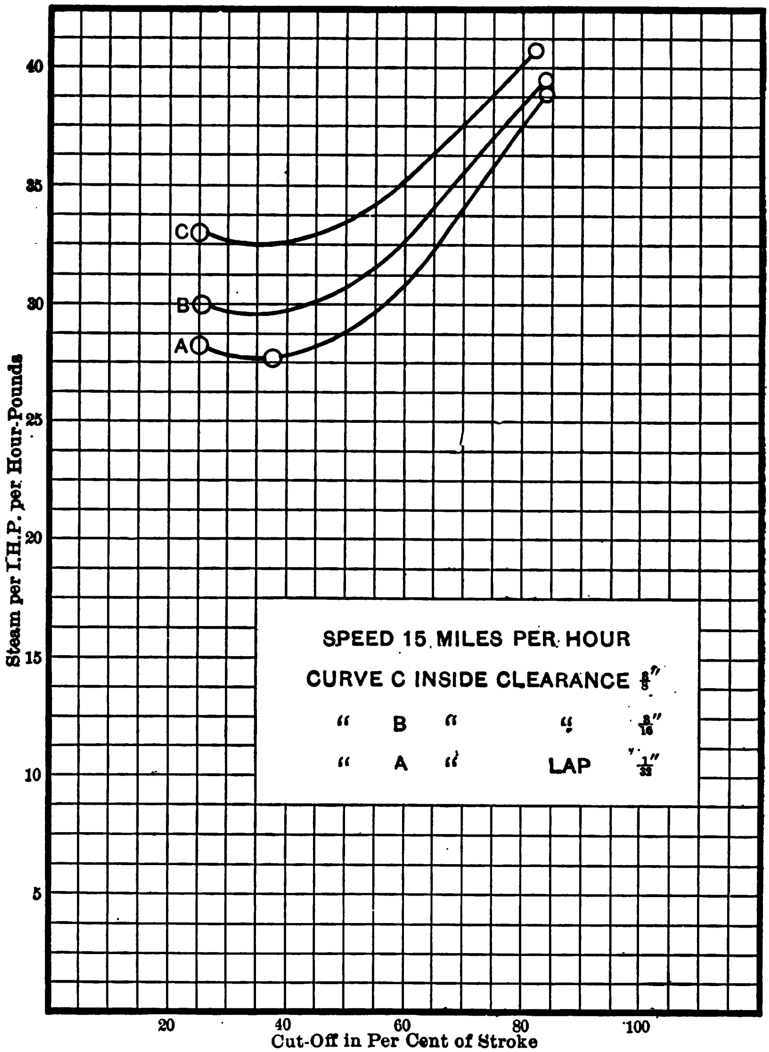


FIG. 190.—Inside Clearance and Steam Consumption.

It has usually been assumed that losses incident to the use of inside clearance are greatest when the cut-off is longest, but the curves of Fig. 190 imply that this is not true. These curves represent the

steam consumption at a constant speed of 15 miles an hour, but with cut-offs varying from 25 to over 80 per cent, and indicate that, as the cut-offs are increased, the steam consumption under all conditions of clearance approaches the same value. This tendency is sufficiently pronounced to justify the conclusion that the losses due to clearance are, other things being equal, diminished as the cut-off is increased.

TABLE LXVII.

STEAM CONSUMPTION AS AFFECTED BY INSIDE CLEARANCE.

Speed, Miles per Hour.	Cut-off, Per Cent of Stroke.	Inside Lap, Inches.	Inside Clearance, Inches.	Steam per H.P.H. Pounds.
		$\frac{1}{32}$	..	33.34
15	80	..	$\frac{3}{16}$	33.92
		..	$\frac{3}{8}$	35.39
		$\frac{1}{32}$	..	28.92
15	25	..	$\frac{3}{16}$	30.03
		..	$\frac{3}{8}$	32.96
		$\frac{1}{32}$	..	26.93
35	25	..	$\frac{3}{16}$	28.9
		..	$\frac{3}{8}$	29.36
		$\frac{1}{32}$	..	30.61
55	25	..	$\frac{3}{16}$	29.72
		..	$\frac{3}{8}$	29.38

The above statement embodies a fair interpretation of the tests under consideration. It would not be safe to generalize therefrom to the extent of assuming that these principles may be applied directly to the performance of all engines, though they may properly be accepted as an indication of what is likely to occur under similar conditions.

In this connection it will also be well to remember that the range of clearance used in the experiments extends far beyond the amounts which have thus far been used in practice. Where inside clearance has been resorted to it has not often exceeded $\frac{1}{8}$ inch, whereas in these experiments a maximum of $\frac{3}{8}$ inch was employed. In view of this fact it is of special interest to note that even with a clearance as great as the latter amount, it ceases to be disadvantageous when the speed of the locomotive reaches the moderate limit of 50 miles an hour.

CHAPTER XVII.

LOCOMOTIVE VALVE-GEARS.*

152. The Function of a Valve-gear.—The valve-gear of a modern locomotive contends with conditions which are difficult to meet. It is designed to so move the valve it drives as to open the port by an amount which, at running cut-off, usually does not exceed three-eighths of an inch, and at speed the entire interval during which any port is open is less than a twentieth of a second. If normal steam distribution is to be maintained the valve must move with great precision, since even a slight change in the extent of its travel, or in the time of its action, becomes relatively large when measured by the small port opening and the brief interval during which the port is open. Moreover, a valve of a modern locomotive, weighing with its yoke 150 pounds or more, requires the application of forces of considerable magnitude to give it motion, and its action involves more than ten reversals in its motion each second. It is necessary that the gear which actuates such a valve be free from lost motion, and that it be so stiff as to admit of no deformation of its parts. With this understanding of the requirement, we shall do well first to examine that type of valve-gear which is common in American practice, and to review briefly its merits and defects.

153. A Stephenson Valve-gear, which is so familiar to all that it need not be illustrated, as designed to drive a flat valve, contains ten joints having motion when the gear is in action, between the axle and the valve-yoke. Lost motion in any of these will modify the valve action. Accurate fitting and the use of case-hardened parts have, however, made the joint of the link motion reasonably satisfactory, but the average gear leaves much to be desired in the way of stiffness. It is true that great progress has been made in this latter respect. The

* Adapted from a paper before the Southern & Southwestern Railway Club, January, 1905.

modern engine does not slow down when the throttle-opening is increased, as used sometimes to be the case with locomotives having bent and limber eccentric blades; but even in the modern locomotive the spring of parts has not been entirely eliminated. This, however, may be greatly reduced by the adoption of the marine type of link, with double hangers, giving support to the link upon both sides, and having connection with a very heavy reverse-shaft and -arms.

154. What the Stephenson Gear Does.—The motion communicated to the valve by the Stephenson gear is one which, beginning from rest, increases to a maximum and then gradually diminishes, until at the end of its travel it comes to rest again. The fact that it approaches its point of rest, and upon reversal recedes therefrom with a relatively slow motion, permits the valve to be controlled in its course more easily than would be the case if its changes in velocity were in obedience to a more complicated law. The gradual change in its rate of motion may well be seen in a curve representing the relative movement of valve and piston (Fig. 191).

In the diagram of this figure, which is sometimes referred to as a valve ellipse, horizontal distances represent piston displacement, and vertical distances valve displacement. The figure shows two ellipses corresponding with two different positions of the reverse-lever. The distance between the mid-position line and the upper portion of either curve indicates the distance which the valve is removed from its central position, and distances along the mid-position line represent piston displacement. Thus, in the original diagram from which Fig. 191 was made, the distance along the line *AB*, from the center line to the curve *c*, is one and three-fourths inches. If we subtract the outside lap of the valve from this distance the result will be the port-opening. For example, in the present case, the outside lap is one and one-half inches, and the port-opening on the line *AB* is one-fourth inch. In the particular case in question the shaded area represents both the portion of the stroke and extent of the port-opening. As many valve ellipses can be made as there are notches in the quadrant holding the reverse-lever. Again, while only the steam-port opening has been referred to, it should be evident, since the displacement of both valve and piston are shown, that the time and extent of the exhaust-port opening can be determined as well as that of the steam-port.

155. Devices for Increasing the Acceleration of the Valve.—The present purpose is less concerned with questions of port-opening

than with the form of the curve representing the motion of the valve, for it will be well at this point to discuss briefly certain modifications of existing gears designed to accelerate the valve motion when the piston is at the end of its stroke. It is the purpose of this class of device to secure a quick and liberal opening of the port, to hold the valve nearly stationary during admission, and to close the port promptly at cut-off. The mechanical arrangements for securing these results usually consist in the addition of mechanism to existing valve-gears. A number of different gears have from time to time been proposed and used. While the means employed vary greatly in different gears the form of the valve diagram will be much the same for all. In contrast with that shown by Fig. 191, it will be a figure having the same

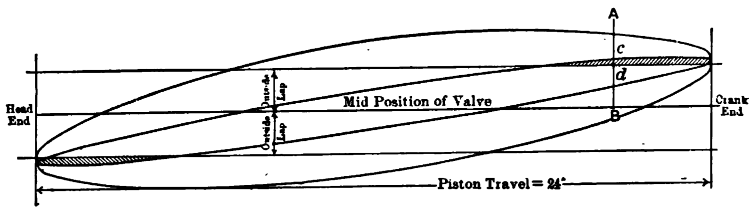


FIG. 191.—Valve-motion Diagram.

total length and the same total width, but with ends much flatter indicating the higher velocities of the valve on either side of its points of reversal. It is evident that increased acceleration requires a gear which is stiffer than that which is usually employed. This fact, together with a knowledge of the care which must be exercised in the construction and maintenance of normal gears, suggests some of the difficulties to be met in designing and maintaining any of these accelerating devices. The conception underlying the accelerated device is good, but the locomotive is an extremely difficult type of engine upon which to employ it. The fact that in locomotive service the valve travel is considerable, and the speeds are oftentimes enormously high, makes it difficult to say just how far a design may be carried to secure such stiffness of gear and increase of wearing surfaces as may be required to perform the work imposed by any of the devices in question.

156. Wiredrawing as a Factor Controlling Valve-gear Design.—Another point to be emphasized in connection with the valve ellipse is the small extent of port-opening. The smaller of the two ellipses shown represents a cut-off of 20 per cent of stroke. The greatest width of the shaded portion on a full-sized diagram is one-fourth inch, which is the maximum steam-port opening at this cut-off.

It is commonly assumed that all wiredrawing in locomotive service is objectionable, whereas, there is one very useful service which it renders. It is that of assisting in the maintenance of uniform conditions upon the boiler while the engine is operating under widely varying conditions of speed. The significance of this statement will appear if we assume an engine at a speed of 20 miles an hour, with reverse-lever and throttle set to create a demand for steam such as will put a fair working load on the boiler. If now, by a change in grade, the load is diminished sufficiently to permit the speed to increase to 40, 50, or even 60 miles an hour, it will generally not be necessary to change the reverse-lever or the throttle, the wiredrawing action coming into play to prevent the cylinder from taking from the boiler more steam than it can supply, the engine meanwhile continuing at all speeds to work at or near its maximum capacity. This automatic regulation of power through wiredrawing is one which means much, both as a matter of convenience in the operation of the locomotive and as a factor affecting boiler repairs.

But there are objections to wiredrawing which are real and sometimes serious, arising from the fact that as the process proceeds the cylinder action becomes less efficient. The loss on this account is, however, smaller than is generally supposed. Its value may be seen in the data of Chapter V., or better, perhaps, in a brief summary of facts derived from a locomotive carrying 200 pounds steam pressure, which is as follows:

Speed	50,	steam per	I.H.P.H., 24.9
"	40,	" "	I.H.P.H., 23.7
"	30,	" "	I.H.P.H., 24.6
"	20,	" "	I.H.P.H., 25.4

These values correspond with the cards of Fig. 192 for the 28 per cent cut-off.

Results for other cut-offs are substantially the same. At 60 miles an hour, were the data available, the consumption would probably rise to 27 pounds. It is a principle in steam engineering that, other things remaining the same, the steam consumption should diminish with increase of speed. This principle would apply in locomotive service were it not for the fact that with each increment of speed there is a change in the distribution of steam. The result is that while the steam consumption diminishes until a speed of 35 or 40 miles an hour is reached, after this point it increases, the losses through

wiredrawing being greater than the thermodynamic gain due to the increase of piston speed. Referring to the figures above, there can be no doubt but that the change from 23.7 pounds to 24.9 pounds is due to wiredrawing. If for this series of tests the steam consumption had been obtained for a speed of 60 miles, it would have shown an increase of two pounds, or, say, 10 per cent above the minimum.

In conclusion, therefore, with reference to wiredrawing with increase of speed, it may be said that its presence is a necessary accompaniment of valves and gears now common; that is, it serves a useful purpose in regulating the demands which the cylinders make upon

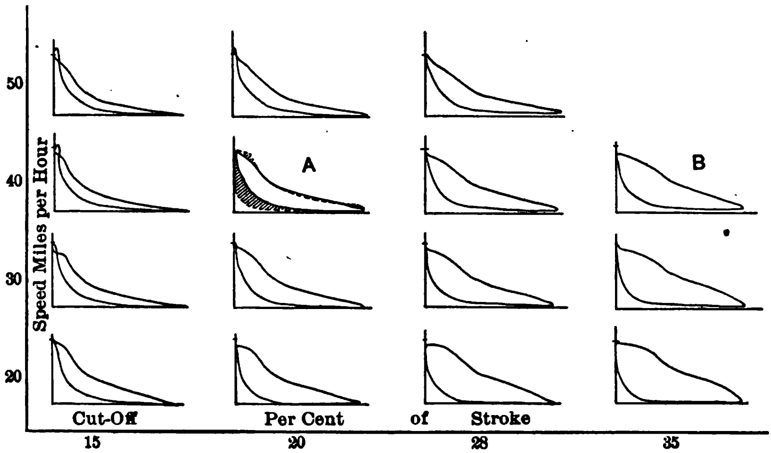


FIG. 192.

the boiler, but, at the highest speeds, it is responsible for some increase in steam consumption.

157. Improved Valve-gears.—Countless devices have been proposed affecting either the valve or the gear which gives it motion, whereby the card may be made larger than that which results from the normal link-driven valve. A typical improved card is shown at *A*, Fig. 192. Concerning such devices it should be noted that it is usually assumed, though the assumption is erroneous, that anything which increases the area of an indicator-card is desirable. For example, in Fig. 192, for 20 per cent cut-off and a speed of 40 miles an hour (card *A*), the plain outline is the normal card around which has been drawn a so-called improved card. The difference is the shaded area and is presumably the result of the adoption of some new form of gear. Obviously, the shaded

area represents increase of power. The first mistake that is made concerning the change is that the increase in power results in no expense. Again, while the truth of the preceding statement may be admitted, it is often urged that one may measure pressure and volume represented by two indicator-cards, such as are shown by Fig. 192, and derive therefrom an estimate of the relative amount of steam used per horse-power per hour under the conditions which each represent. Such estimates are, in fact, fairly reliable when made between cards agreeing closely in form, and when all conditions of running are the same, but as a general proposition nothing is more misleading. If there are differences in speed, or in initial or final pressure, or in the number of expansions, the percentage of the total amount of steam used, which is shown by the indicator, will change. Anything which may produce a change in the temperature of the metal of the cylinder at any one point in its cycle is likely to produce changes in the whole cycle. As is well known, a considerable percentage of the steam drawn from the boiler for each stroke of the engine condenses on entering the cylinder. While the interchange of heat causes some change in the amount of water in the cylinder as the piston proceeds on its course, by far the larger part of the initial condensation continues in the cylinder until the exhaust-port is open, when it flashes into steam and disappears with the exhaust. While the process is a complicated one, and cannot within the limits of this chapter be accurately defined, the fact is that any change in the form of any line bounding an indicator-card has its effect upon the amount of steam which must be admitted to make up the loss due to initial condensation. A change in the cycle remote from the period of admission may have as pronounced an effect on the quantity of steam required as a change occurring during the period of admission itself. There is, in fact, no way to measure the performance of a steam-engine but by a process which determines the actual weight of the feed or the exhaust. Again, a further illustration of the fact that a mere increase in the area of an indicator-card is not significant is to be found in the ease with which such increase of area may be secured. In locomotive practice it is quite unnecessary to adopt a new gear. If. under the conditions prescribed, the normal card *A* at 20 per cent cut-off (Fig. 192) is not large enough for the work, the reverse-lever may be advanced on its quadrant until the cut-off is 35 per cent, whereupon, in this particular case, the normal card *B* becomes equal in size with the card representing an assumed im-

proved gear. The real questions, therefore, may generally be stated as follows: Is the improved card at 20 per cent a better card than the normal card of equal area at 35 per cent cut-off? Will the former yield a horse-power upon the expenditure of less steam than the latter? It is upon this latter statement that the argument rests. No device which seeks to improve the steam distribution in a locomotive can succeed which does not save steam when compared with devices previously in use. In proportion as it saves steam it both increases the efficiency of the engines and increases their maximum power, for since the boiler capacity is limited a pound of steam saved is a pound of steam available for additional services.

Turning now to a consideration of the margin upon which those who would improve valve-gears have to work, it must be admitted that it is not large. Results have already been quoted which prove that the locomotive with all its wiredrawing gives a horse-power in return for less than 24 pounds of steam per hour. This is near the minimum. From this performance of a simple locomotive having a normal valve-gear, with its narrow port openings and wiredrawing effects, we may turn to the Corliss engine, the action of which is generally accepted by all improvers of locomotive valve-gears as a standard of perfection. Such an engine, with its large port-opening and its prompt movement of the valves, can in fact be relied upon to give as good a performance as engines having any other type of valve-gear operating under similar conditions of speed and pressure. Corliss engines, having cylinders which are comparable in size with those of locomotives, and which work under a similar range of pressure, are, however, not common, and hence it is not easy to command data for the proposed comparison. Generally, simple Corliss engines work under a lower pressure than locomotives. A good performance for a simple Corliss engine exhausting into the atmosphere is that of an 18×48 Harris-Corliss engine, for which the steam consumption was 23.9 pounds per hour.* The steam pressure supplied this engine was only 96 pounds by gauge. On the basis given the engine should, when supplied with steam at 180 pounds, which is the pressure under which the locomotive data were obtained, require less than 23 pounds of steam per horse-power per hour. Straining the facts applying to the two classes of engines as widely apart as a knowledge of existing data will possibly permit, we may assume that a Corliss engine, if given the advan-

* Peabody's Thermodynamics.

tage of the high steam pressure and high piston-speed common in locomotive service, may give a horse-power hour on the consumption of two pounds less of steam, or approximately 8 per cent less than the locomotive. This, then, is the margin upon which those who seek to improve the locomotive valve-gear must expect to work. While it is well worth attention, it cannot revolutionize practice.

158. Foreign Valve-gears.—Having now considered the defects and merits of the link motion, we may inquire concerning other forms

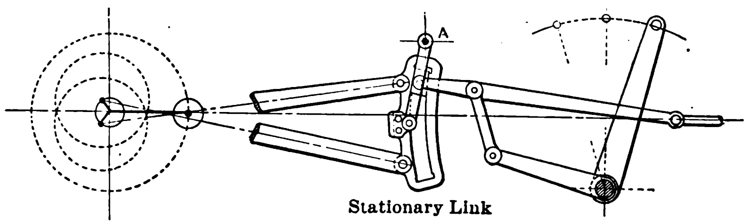


FIG. 193.

of gears which, in foreign countries at least, are in common use. In England one occasionally sees the stationary link, as shown by Fig. 193, the link-hanger of which is suspended from a fixed pin, *A*, while the reverse-shaft is connected with a radius-rod which communicates the motion of the link to the valve-spindle. A modification of this gear is found in the straight or Allen link, shown by Fig. 194, in

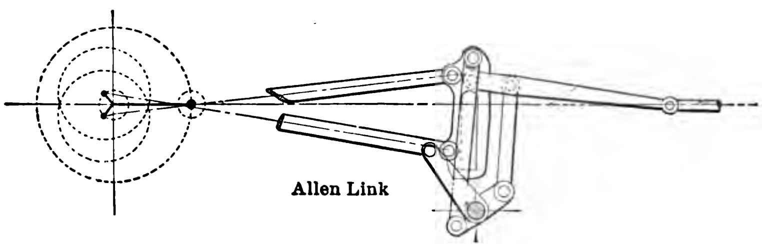
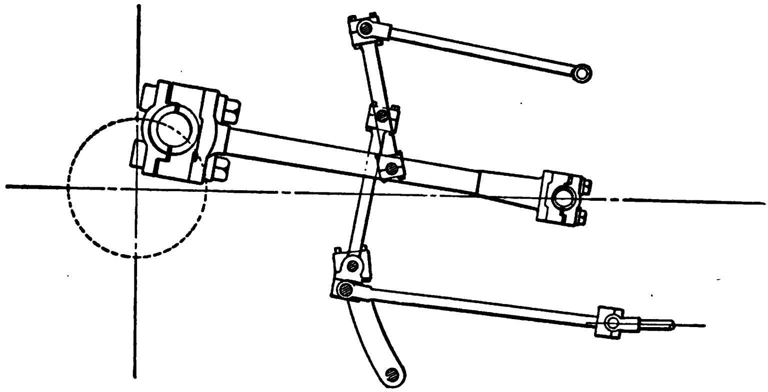


FIG. 194.

which both the link and the radius-rod have connection with the reverse-shaft, which in reversing causes them to move in opposite directions.

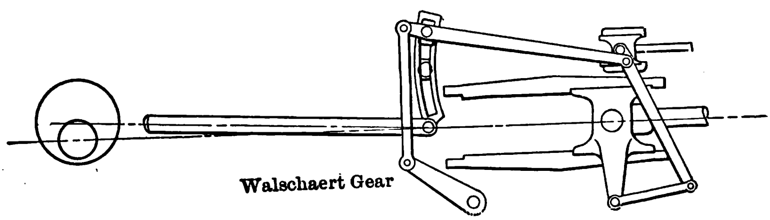
The Joy gear (Fig. 195), which has been, and still is, extensively used in English practice, requires no eccentric, but receives its motion directly from the main rod. The reversal of the engine and changes

in the travel of the valve are, in this gear, accomplished by varying the inclination of the curved guide along which a block is forced to travel with each reciprocation of the main rod. The guide remains stationary, except as acted upon by the reverse-lever.



Joy Gear
FIG. 195.

As a last example from foreign practice is presented the typical gear of Continental Europe, namely, the Walschaert gear (Fig. 196). In this gear motion is derived from the cross-head and from a single eccentric, or, where the gear is outside of the frame, from a return-crank. These motions are so combined, by means of a system of



Walschaert Gear

FIG. 196.

connected rods and a link, as to give the usual motion to the valve. A radius-rod, connecting with the link-block, receives motion from the reverse-shaft in reversing and in changing the travel of the valve.

The fact concerning these foreign gears which is to be emphasized is that they all give a motion to the valve, which may be represented by a valve ellipse, such as that shown by Fig. 191. There are minor differences in the character of the resultant motion

which for the present purpose need not be discussed, the important fact being that the mechanisms in question are not to be regarded as reform gears. They are not to be contrasted with the Stephenson link motion, but compared with it. They do not exist because they give a better steam distribution than is obtained from the link motion of this country, and, except as may be hereinafter suggested, they are not superior to the gears common in this country. With this understanding of the matter, we may now examine the several gears referred to, and seek to find a reason for their existence in the conditions under which they are used.

159. Adaptability of Valve-gears.—From that which has already been stated, it may be surmised that the Stephenson link motion, as used upon American locomotives, constitutes a convenient and acceptable gear. There are at least four good reasons why this is so, which may be enumerated as follows:

1. The gear gives a satisfactory distribution of steam.
2. Its design readily adapts itself to conditions involving the use of a rocker between the axle and the valve, and the rocker has until recently been regarded as an essential element in the design of the American locomotive.
3. The accessibility of the forward driving-axle, and the unoccupied space between the frames in American locomotives as a point of attachment for eccentric, has supplied ideal conditions under which to develop the link motion.
4. Other forms of gear which have been used elsewhere have been thought impracticable in American service, because, owing to the small diameter of driving-wheels, which, until recently, have prevailed in this country, the mechanism of such gears often extends too near the road-bed for safety.

The stationary link (Fig. 193) has proven attractive to the designers of many English engines, because it involves the use of a single eccentric, room for which is more easily found upon the inside connected engines common to English practice, where the greater part of the forward axle is occupied by the cranks, than for the two eccentrics common in American practice.

The Allen link responds to conditions similar to those which are met by the stationary link, but is a better design, since by it the lines of motion are kept much nearer the lines of force.

The Joy valve-gear is distinctively English in its design and in the extent of its use upon locomotives. With the increased size of the inside

connected locomotive, the inside crank of the forward axle gradually developed proportions which made it difficult to find room even for a single eccentric. Confronted with these conditions, the Joy gear naturally makes a strong appeal to the designer. This gear takes its motion from a connecting-rod, and leaves the forward driving-axle to be occupied wholly by the cranks. For locomotive service the Joy gear is perhaps inferior to either of the other motions described, since irregularities in the track materially affect the distribution of steam. For example, when a low joint permits the wheel to drop, the connecting-rod partakes of its motion and carries it on to the valve-gear and to the valve.

The Walschaert gear, which is extensively used in Continental Europe, may be either inside or outside of the frames of the engine. In European practice, where inside cylinders have been much employed, it has generally been outside of the frames. Its design makes a strong appeal to the designer who is forced to go outside of the frame with his valve motion. The gear is one in which the metal for the parts may be well distributed to transmit the stresses which are brought upon them, and for this reason it may constitute a very stiff gear, though its individual parts are relatively very light. For these reasons its use is likely to increase in American practice.

160. **The Conclusion** to be drawn from the foregoing discussion should not be one of discouragement for those who are interested in improving the locomotive valve-gear. Forms of gears now common have been commended, but this does not imply that better ones may not be devised. The argument is, however, that real and lasting improvement is to be looked for more along mechanical lines than in attempts to improve the character of the motion imparted to the valve. While there is a chance for slight saving in fuel, the real economy which may result from the adoption of a better gear is to be found in its lower maintenance cost, and in the greater certainty of its action, rather than in pounds of coal saved. What is most desired in a valve-gear is a mechanism which, under the adverse conditions of actual locomotive service, will give to the valve that precision of movement it is designed to have. Along these lines there is ample opportunity to improve present practice.

CHAPTER XVIII.

ACTION OF THE COUNTERBALANCE.*

161. The Problem of Balancing.—In the mechanism of a locomotive the revolving parts at the crank-pins, together with the reciprocating parts connected therewith, are balanced more or less completely by the addition of masses, or counterweights, to the drivers. But since the counterweights move in circular paths, it is only the horizontal component of the radial force derived from them which can serve to neutralize the effect of the reciprocating parts; the vertical component of all that portion of the force which applies to reciprocating parts is unbalanced. This unbalanced vertical component causes the pressure of the driver on the rail to vary with every revolution. Whenever the speed is high it is of considerable magnitude, and its change in direction is so rapid that the resulting effect upon the rail is not inappropriately called a hammer-blow. Many practical demonstrations have been had of the magnitude of the forces involved. Heavy rails have been kinked, and bridges have been shaken to their fall, all under the action of heavily balanced drivers revolving at high speeds.

Such results do not ordinarily follow from the operation of the locomotive, and indeed cannot do so if the locomotive is properly designed and its speed is confined to reasonable limits, but the value of the vertical component of the counterbalance of a modern locomotive is nevertheless of considerable magnitude. This may be judged from the fact that the reciprocating parts on each side of such an engine exceed 1000 pounds in weight, and that the drive-wheels with which they are connected not infrequently turn 300 revolutions a minute.

* The facts of this chapter were presented at the New York meeting (December, 1894) of the American Society of Mechanical Engineers, forming part of Vol. XVI of the Transactions.

Various attempts have been made to so dispose the reciprocating parts of a locomotive that, by their inter-action, a satisfactory degree of balance may be secured. The de Glehn balanced compound constitutes a well-known though not the only example of this type, but such engines have not yet come into general use in American practice, nor are they likely soon to do so. In dealing with the mechanism now common to American locomotives the most which the designer has been able to accomplish in the matter of balancing is in the nature of a compromise. It is found that the mass of the locomotive is sufficient to absorb a portion of the force set up by the motion of the reciprocating parts, and that, as a consequence, the counterweights put in the wheels need not be sufficient entirely to balance these parts. Obviously, the less the weight which is put into the wheel the smaller will be the vertical component which constitutes the objectionable result arising therefrom. An old rule was to provide balance in the wheel for from 75 to 80 per cent of the weight of the reciprocating parts. A later rule of the Master Mechanics' Association provides that $\frac{1}{400}$ part of the weight of the locomotive may be subtracted from the weight of the reciprocating parts, the difference to be the weight to be balanced. Engines designed in accord with these rules are so well balanced horizontally that they do not impart vibrations to the trains they pull, while the excess weight of the counterbalance is not sufficient to prove destructive to tracks or structures.

The forces which are brought into action by the presence of the counterbalance have been elaborately studied, and their precise effect upon the pressure of contact between wheel and rail have at times been the subject of much discussion.

The present chapter describes a series of experiments which were undertaken at the Engineering Laboratory of Purdue University to demonstrate the varying pressure between the revolving driver and the rail of the track. The essential features of these experiments consisted in passing a soft iron wire of small diameter under the moving wheel. It was expected that the varying thickness of the wire, which had been subjected to this process, would show the effect of variation in pressure between the wheel and the track. If the wheel should leave the track entirely a portion of the wire would retain its full diameter; and the real purpose of the experiments, as originally planned, was to determine whether at any speed easily attained the driver would actually rise from the track.

162. **Experimental Method.**—The apparatus employed consisted chiefly of the locomotive Schenectady mounted upon the Purdue

testing-plant. To guide the wire, which was to be fed under the driver, a length of $\frac{3}{8}$-inch gas-pipe was secured to the laboratory floor in front of each driver included in the experiment (Fig. 197). Three pipes were thus arranged. A deflector plate was fixed behind the main driver to turn the wire delivered by this wheel away from the rear driver; but, except for this plate, no attempt was made to control the course of the wire after it left the wheel. The wire was of common annealed iron 0.037 inch in diameter. It was prepared by being carefully straightened and cut into lengths of twenty feet; that is, about 3.5 feet longer than the circumference of the drivers, and two incheslonger than the guide-pipe, in which the lengths were to be fed to the wheels. Wires thus prepared were laid in light wooden troughs to preserve them from injury, and a trough thus supplied was placed in line with each guide-pipe (Fig. 197). In conducting the experiments an

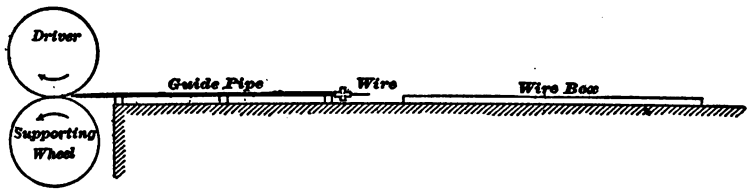


FIG. 197.

operator at each pipe drew a wire from the trough and passed it into the pipe, until only about two inches of the length remained outside. From the relative length of guide-tube and wire it was known that the opposite end of the latter was now close to the driver. When desired conditions of speed had been secured and a signal given, a touch of the operator's finger upon the end of the wire was sufficient to start the opposite end under the wheel. The starting of the wire was accomplished without commotion. The man in charge was conscious only of having touched it. To an observer who watched for the wire as it came from the driver it gave the impression of a quivering beam of light, which an instant later became a loosely tangled thread of metal. Or, if one kept his eye upon the wall of the laboratory, against which the wire was allowed to impinge, he saw the whole tangled coil appear instantaneously, and without apparent cause. The initial end of each wire was, in plan, of the outline shown by Fig. 198, from which it would appear that when the wire came under the influence of the wheel's motion the tensional stress upon sections near the end, as at A, exceeded the elastic limit of the material, this stress being required to impart motion to the mass of wire to the

right of *A*. The weight of the twenty-foot length was about one ounce, and the time occupied in its passage was usually a fifth of a second. These facts will help to show the significance of the speeds used in the experiments.

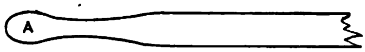


Fig. 198.

The speed of the locomotive was noted from a registering-counter, and also by a Boyer speed-recorder, a permanent record being obtained from the latter instrument. To assist in connecting the effect produced on the wire with definite phases of the wheel's motion a nick was made with a sharp chisel across the face of each driver, in line with the counterweight, as at *A* (Fig. 199). An impression of this nick was sharply defined upon every wire that passed under it. The initial end of the wire could, as has been stated, be determined by an examination; but to leave no doubt as to this matter, and for the purpose of giving a second reference point, one of the wheels was marked with two parallel lines ninety degrees from the first reference line, as at *C* (Fig. 199).

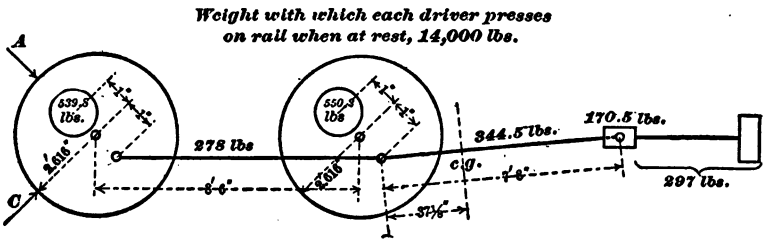


Fig. 199.

It was found by a comparison of reference marks that distances along the length of the wires could be taken as representing equal distances around the face of the wheel. Thus, the length of each wire being greater than the circumference of the wheel, it would sometimes happen that a single wire would receive two impressions from the same reference mark, the distance between the two points thus impressed upon the wire being found equal to the circumference of the wheel. This fact made it easy to connect effects left upon a wire with the wheel positions (crank-angles) producing them.

Many of the wires produced by the experiment described have

since been carefully calipered at five-inch intervals, the results plotted, and a smooth curve drawn through the points thus located. Some of the results thus obtained are presented as Figs. 200, 201, and 202,

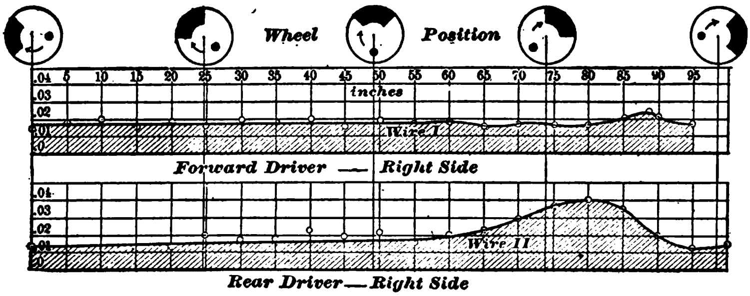


FIG. 200.

the points representing the actual thickness of the wires being designated by means of small circles. It will be seen that all diagrams are plotted with reference to definite wheel positions.

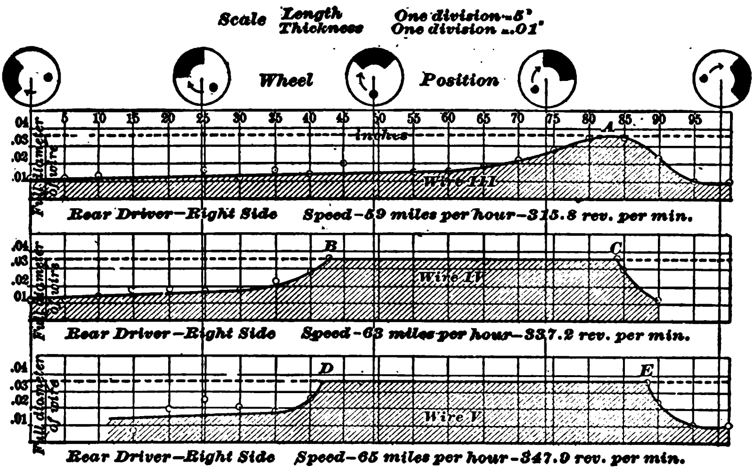


FIG. 201.

163. The Balance of the Locomotive.—Before attempting a discussion of results in detail, it is necessary to consider somewhat briefly the condition of balance of the locomotive experimented upon. The engine, as delivered by its builders, was balanced for the road; but to increase its steadiness in the laboratory weights were afterward

added in equal amounts to the several wheels, until a *full horizontal balance* had been secured.* The revolving and reciprocating parts which required counterbalancing, exclusive of the crank-pins and

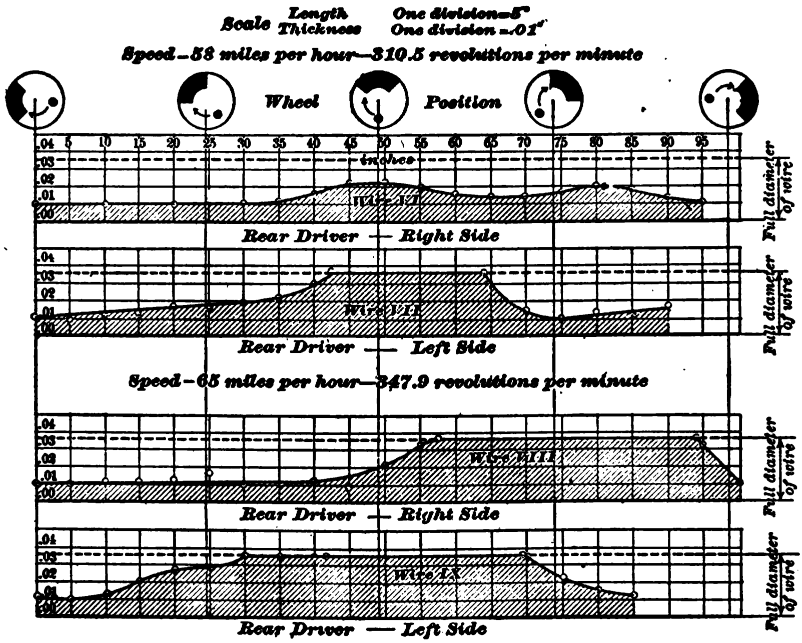


FIG. 202.

crank-pin bosses, which are assumed to be parts of the wheels themselves, were found to weigh as follows:

Piston and piston-rod	297.0 lbs.
Cross-head with part of indicator rigging attached	170.5 "
Main rod	344.5 "
Side rod	278.0 "
Total for one side	1090.0 lbs.

For complete horizontal balance it was required that the sum of the weights, making up the counterbalance of the two wheels on the side of the engine under consideration, should be equivalent to 1090.0 pounds acting at a radius of one foot. To ascertain the distribution

* On January 23, 1894, the plant from which the results herein described were obtained was destroyed by fire. The new plant, now in operation, does not require the locomotive to be in complete horizontal balance.

of balance between the wheels it was necessary to examine them separately. Calculations based upon drawings of the wheel centers gave the following results:

	Main Wheel.	Rear Wheel.
Balance cast in rim and between the arms, plus the weights added at the laboratory, all reduced to equivalent weights acting at a radius of 12 inches	744.1	725.7
Weight of crank-pin and crank-pin hub to be subtracted	187.1	179.1
Net weights available to balance revolving and reciprocating parts acting upon the crank-pins	557.0	546.6

The sum of the net weights thus obtained for both wheels (1103.6 pounds) is 13.6 pounds greater than the sum of the actual weights to be balanced. The engine is known to have been in perfect horizontal balance, the experimental methods adopted in securing this condition serving to indicate when the weights were changed even to the extent of a single pound. The calculated weight in each wheel is, therefore, assumed to be $\frac{13.6}{2}=6.8$ pounds heavier than the weights themselves, and this amount has been subtracted as a correction from the net weights given above, making the

	Main Wheel.	Rear Wheel
Corrected net weight of counterbalance, available to balance revolving and reciprocating parts, acting upon the crank-pins	550.2	539.8

The weights of the parts involved, together with certain dimensions, are summarized in Fig. 199.

Taking the weights of side rod and of main rod, as already given, and considering 0.6 of the weight of the latter as a revolving part,

	Main Wheel.	Rear Wheel.
The excess of balance over that required for revolving parts alone is	204.5	400.8

which shows 66 per cent of the balance for reciprocating parts to be in the rear wheel.

Six different rules for balancing locomotives for the road, reported as being in common use, give weights of counterbalance for the locomotive in question, as follows:

	Main Driver.	Rear Driver.
Rule A (for freight-engines only)	467	260
" B (for all classes of service)	462	322
" C " " " " "	547	340
" D " " " " "	570	340
" E " " " " "	573	366
" F " " " " "	588	381
Average of five rules from B to F inclusive	548	350

Compared with these several standards the weights of the counterbalances in the Purdue engine stand as follows:

	Main Wheel.	Rear Wheel.
By Rule A (for freight service only)	17.8% too heavy	107.6% too heavy
" " B (for all classes of service)	19.1% " "	67.6% " "
" " C " " " " "	0.6% " "	56.9% " "
" " D " " " " "	3.5% too light	56.9% " "
" " E " " " " "	4.0% " "	47.5% " "
" " F " " " " "	6.4% " "	41.6% " "
By the average of five rules from B to F inclusive..................	0.4% too heavy	54.2% " "

It is evident, therefore, that the weight of the counterbalance in the rear wheel, from which most of the results about to be discussed were obtained, is in excess of that allowed by good practice, as expressed by the rules already given. But practice cannot always conform to the law by which it assumes to be governed. It often happens where wheels are of small diameter, and the connections are heavy, as in mogul or consolidation engines, that there is not sufficient room in the main wheel to get in a counterbalance large enough for the revolving parts alone; in this case, therefore, the balance for reciprocating parts of this wheel must be taken by the other coupled wheels, in addition to that which, under the rules, would be counted as properly belonging to them. By this process wheels having revolving parts, which are relatively light, are employed to balance a larger per cent of all the reciprocating weights. Again, almost any eight-wheeled engine, balanced in an approved manner, will, if the coupling-rod is removed, have an excess of balance in the rear wheel greater than that for the engine under consideration; and such engines are not infrequently run while disconnected.

These considerations will serve to show that while the total weight of the counterbalances of the Purdue engine is, for reasons already stated, heavier than would be considered necessary for the road, and while at the time of the experiments the weights were not well distributed between the wheels, yet the conditions which existed are not at all rare. Doubtless many wheels are running which carry a greater counterbalance, when compared with the revolving weights to be balanced, than did the rear wheel of the Purdue locomotive.

164. Results.—Attention has already been directed to the fact that, in the engine experimented upon, the excess of weight in the counterbalance over that required for the revolving parts alone was much greater for the rear driver than for the main driver. As the

lifting effect is proportional to this excess of weight, it follows that wires run under the rear driver were likely to show more variation in thickness than those under the main driver. Results of experiments upon this point are shown by Fig. 200, which represents wires obtained at the same instant from the main driver and the rear driver respectively. It will be seen that the wire (*I*), from the main driver, shows but slight variation in thickness, notwithstanding the high speed (312 revolutions per minute), and it may be said that no wire was ever obtained from this wheel which gave evidence that the wheel had left the track. From mathematical considerations it can be shown that this wheel would not be expected to lift at speeds below 80 miles per hour (428 revolutions per minute), and such speeds are not practicable with wheels of the diameter experimented upon.

Passing now to an inspection of wire *II* (Fig. 200), from the rear wheel, which was obtained at the same instant with wire *I*, it will be seen that there is a jump of the wheel just after the counterbalance has passed its highest point, which, when compared with the corresponding movement of the main driver, is very pronounced. Wires from this wheel at higher speeds are shown by Fig. 201. In this figure the full diameter of the wires is in each case shown by a dotted line drawn parallel with the base line. Wire *III*, made at 59 miles (316 revolutions), shows that there was an instant in the passage of the wire, corresponding to the point *A*, when it was barely touched by the wheel. Increasing the speed to 63 miles (337 revolutions) increased the lifting action of the wheel to the extent shown by wire *IV* (Fig. 201). At the point *B* the wheel parted contact with this wire, and did not again touch it until the point *C* was reached, an interval of about 40 inches, the portion of the wire between *B* and *C* being entirely round and apparently unaffected by its passage under the wheel. A further increase of speed gave, as is shown by wire *V*, a still greater length of full wire, the distance from *D* to *E* being very nearly equivalent to a quarter-revolution of the driver.

It will be seen that all of these wires (*II* to *V*, Figs. 200 and 201) substantially agree in showing the maximum lifting effect to occur after the counterbalance has passed its highest point, an effect undoubtedly due to the inertia of the mass to be moved; also in showing that the rise of the wheel from the track is more gradual than its descent. The latter condition follows as a sequence of the first.

Portions of the wires not shown on the diagrams do not vary much in thickness. The metal is rolled so thin by the normal pressure

of the wheel that further increments of pressure do not greatly affect it. The wires, therefore, do not emphasize the destructive effect of the variation of wheel pressure when the change is insufficient to lift the wheel from the track.

It now remains to mention the effect of certain disturbing elements, which are shown by the experiments, to modify the actual movement of the wheel, other conditions remaining constant. For the rear wheel these disturbing elements are all in the nature of vibrations. The first to be noticed is the rocking of the engine upon its springs, which motion tends to vary the pressure of the wheel upon the track independently of the action of the counterbalance. At one revolution the effect of the rocking may oppose the action of the counterbalance, and at the next revolution it may supplement the action of the counterbalance in producing a vertical movement of the driver. Again, the effect of the rocking may at a given instant be *nil*, and the wheel may rise under the action of the counterbalance; but in another instant the effect of the rocking appears, and the path of the wheel while in the air is modified and its time of descent changed. Thus, the existence of this vibration makes it impossible to duplicate wires with certainty, even though the speed is constant. Its effect is well shown by Fig. 202. Wires *VI* and *VII* were taken from the rear drivers at the same instant, one from the right side, the other from the left; the speed, therefore, must have been the same for both. The right driver lacked a good deal of leaving its wire; but the left driver was in the air for a tenth of a revolution. Again, wires *VIII* and *IX* were made in the same way at a higher speed; and here, while both drivers were off the track, the results are reversed, the right driver giving the greater length of full wire. It will also be seen from the diagrams that not only is the extent of the vertical movement of the driver modified by the rocking of the engine, but the position of the wheel when such motion occurs is changed. It is evident, therefore, that this movement of the engine upon its springs will prove a serious difficulty whenever an attempt is made to predict as to the precise movement of the center of gravity of the driver, whether the method of investigation be mathematical or experimental.

There appears, also, to be a vibration of parts, as, for example, of the wheel as a whole, these vibrations being of small amplitude. Evidence of the presence of such vibration is shown by the location of points on the diagrams of wires, Figs. 200 to 202, which points represent the thickness of the wires as found by measurement. Referring

especially to wires *I* and *II* (Fig. 200) it will be seen that the actual thickness of the wire alternately increases and diminishes with every point. The time involved in passing from one high point to another (a distance of ten inches) was about 0.01 of a second. This vibration may be traced on other diagrams; its amplitude is from two to four thousandths of an inch only. Whether the process of introducing the wire starts, or has any connection with this vibration, the experiment does not show.

A third class of vibrations is made apparent by a duplication upon the wire of the reference mark on the wheel. As has been stated a light nick from a sharp chisel was made across the face of the wheel to serve as a reference mark. This nick leaves a clear-cut projection upon the wire. But at high speeds the single nick across the face of the wheel leaves two projections upon the wire, showing that after making one impression the surface of the wheel must for an instant have actually cleared the wire and then impressed itself a second time. The distance between these projections on the wires varies somewhat, but is usually about an eighth of an inch, which represents a time interval between the two impressions of about 0.008 of a second. The contact between wheel and track is, therefore, not continuous, but is a succession of exceedingly rapid impacts. These vibrations cannot affect the wheel as a whole; they are doubtless due to the elasticity of the materials, and involve only the parts immediately about the point of contact.

165. Conclusions.—The results of the experiments appear to justify the following conclusions:

1. Wheels balanced according to usual rules (which require all revolving parts, and from 40 to 80 per cent of all reciprocating parts, to be balanced, the counterbalance for the reciprocating parts to be distributed equally among the several wheels connected) are not likely to leave the track through the action of the counterbalance, and cannot do so unless the speed is excessive.

2. A wheel which, when at rest, presses upon the rail with a force of 14,000 pounds, and which carries a counterbalance 400 pounds in excess of that required for its revolving parts alone, may be expected to leave the track through the action of the counterbalance whenever its speed exceeds 310 revolutions per minute.

3. When a wheel is lifted, through the action of its counterbalance, its rise is comparatively slow, and its descent rapid. The maximum lift occurs after the counterbalance has passed its highest point.

4. The rocking of the engine on its springs may assist or oppose the action of the counterbalance in lifting the wheel. It therefore constitutes a serious obstacle in the way of any study of the precise movement of the wheel.

5. The contact of the moving wheel with the track is not continuous, even for those portions of the revolution where the pressure is greatest, but is a rapid succession of impacts.

CHAPTER XIX.

MACHINE FRICTION.

166. A Statement of the Problem.—The power which is developed in the cylinders of a locomotive is the sum of that which is delivered at the draw-bar and the losses which occur between the cylinders and the draw-bar. These losses, in the case of a locomotive upon the road, include the frictional resistance of the machinery, the resistance of the atmosphere, and the rolling and journal friction of the locomotive-truck and tender. Upon a testing-plant the only loss which occurs is that due to the factor first named. An interesting problem, therefore, which at once presented itself with the advent of the locomotive testing-plant, was a determination of the amount of power absorbed by the machinery under different conditions of running. There had been no opportunity previous to the establishment of the Purdue plant to make such a determination in connection with locomotives, and even in stationary practice the difficulties attending the measurement of high power when delivered from a fly-wheel has operated to limit available data to plants which are comparatively small in size.

The process which must be followed to determine the machine friction of any locomotive under load is one requiring accurate observations. This grows out of the fact that the value sought appears as a difference between two quantities which are relatively large. The two quantities involved are cylinder power and draw-bar power. The value of these may be ten times as great as their difference, and hence an error of one per cent in a determination of the indicated power, or of the draw-bar stress, may mean an error of ten per cent in the value of the difference which is the factor sought. It was with a full appreciation of the significance of this statement that the experiments to determine the machine friction of a locomotive were undertaken.

167. Methods.—Every efficiency test of the Purdue locomotive is expected to yield all data necessary to a determination of machine friction. The present discussion, however, deals chiefly with tests which involved a short run designed to give in a few minutes' time such data only as is required to determine the frictional loss. Each such test was run under prescribed conditions of speed, steam pressure, cut-off, and throttle-opening. The engine having been warmed by preliminary running, was brought under the conditions prescribed for the tests, after which, upon signal, all observations were simultaneously taken. These observations were four times repeated at intervals of four minutes, during which time the operation of the engine was maintained as nearly constant as possible. Each test, therefore, yielde 16 indicator-cards and such observed data as are given in the exhibit of observed results on page 335. This exhibit is a typical data sheet for such a test.

In the conduct of this work four indicators were used, one of which was closely connected to each end of each cylinder. An absolute drum motion, connected by short cords, was in regular use upon the locomotive. Great care was employed throughout the tests in question in keeping the indicators in perfect order, in working up the cards, and in checking numerical calculations based thereon. Because of these precautions it was felt that the indicated power of the engine was obtained with a degree of refinement not often equalled in locomotive work.

A correct measurement of the power delivered at the draw-bar was found to be a matter of greater difficulty. Accuracy at this point involved not only the design of a traction dynamometer, but also a study of the effect of the mounting mechanism upon the locomotive as a whole. So serious were these difficulties that several years passed before accuracy was obtained in measuring the actual tractive power exerted by the locomotive. A brief description of the nature of these difficulties will be of interest.

168. Difficulties Encountered in Measuring Draw-bar Stresses.—When the experimental locomotive was first installed in 1891, it was provided with a traction dynamometer made up of a very simple system of levers, which required a dash-pot of considerable size to steady them. It was soon found that the delicacy of this apparatus was insufficient to permit a high degree of accuracy in the determination of draw-bar stresses. In consequence of this conviction, when, in the rebuilding of the laboratory after the disastrous fire of 1894,

RUNNING CONDITIONS.

Speed, miles per hour (approximate)	25
Steam pressure (approximate)	135
Reverse-lever, notch from center forward	3
Cut-off, per cent of stroke	45
Throttle	Wide open.

OBSERVED RESULTS.

Gong.	Time of Gong.	Revolution Counter.			Pressure.		Dynamometer.	Engine Position.
		30 Seconds Before.	30 Seconds After.	Difference.	Boiler.	Dry-pipe.		
1	3.20	2097	2223	126	136	132	5840	+1.5
2	3.24	2637	2760	123	126	124	5850	+1.5
3	3.28	3154	3304	150	126	124	5690	+1.5
4	3.32	3716	3837	121	128	125	5860	+1.5
Totals		....	....	520	516	505	23240	+6
Averages		....	....	130	129	126	5810	+1.5

DERIVED RESULTS.

Mean effective pressure (each value the average of four cards).	
Right head end	57.03
" crank end	57.11
Left head end	61.69
" crank end	61.33
Indicated horse-power.	
Right head end	102.28
" crank end	99.25
Left head end	110.70
" crank end	106.18
Total	418.41
Friction, horse-power.	
Total I.H.P.	418.41
Dynamometer horse-power = dynamometer pull $\times$ R.P.M. $\times$ .0004956	374.32*
Frictional horse-power	44.09*
Frictional resistance.	
Mean effective pressure equivalent to friction = F.H.P. $\div$ (.0542 $\times$ R.P.M.)	6.25*
Draw-bar pull equivalent to friction = M.E.P. $\times$ 109.419	683.8*

* No correction applied for engine position.

the opportunity came to improve the plant, it was determined to supply a dynamometer of high quality. An order was subsequently given to the William Sellers Company for an Emery machine, consisting of a hydraulic support, suitably mounted to receive the draw-bar, and a separate scale-case, all of which was duly received, and has since been used. This machine, while capable of measuring stresses as high as 30,000 pounds, is so sensitive that the pressure of one's finger upon the front bolster of the locomotive will produce sufficient motion in the whole mass of the locomotive to give an indication upon the dynamometer scale. Having possession of so delicate an instrument, it was with high expectation that tests to determine the engine friction were entered upon early in October of the first year of the reëstablished plant.

After carrying out a rather elaborate series of tests upon the plan already outlined, it was found that the results were unsatisfactory in that they appeared to follow no law, and because many of the values obtained indicated negative friction, the draw-bar power being greater than that of the cylinders. In entering upon the work at the beginning of a new school year in the fall of 1895, it was decided that the failure of the previous October had been due to the faulty position of the locomotive drivers upon their supporting wheels, and hence the true position of the drivers upon the supporting wheels, both when the engine was at rest and in motion, was first determined.

The importance of this will be seen from the following considerations: With the locomotive at rest, and so located that a line joining the center of each driver with the center of its supporting wheel is vertical (Fig. 203), no force need be exerted along the line of the

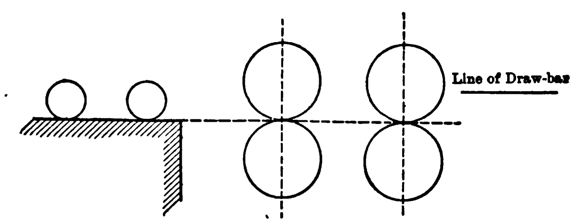


FIG. 203.

draw-bar to hold the engine in its position upon its supporting wheels. Moreover, with the engine in this position and in motion the indication of the draw-bar dynamometer should serve as a correct measure of the force transmitted from the drivers to the supporting wheels,

that is, of the tractive force exerted by the locomotive. In the event, however, that the drivers are ahead of their neutral position on the supporting wheels, or back of the same (Fig. 204), it is evident that when the engine is at rest force must be exerted along the line of the draw-bar to hold it in position, and that with the engine in motion in either of these positions the indication of the draw-bar dynamometer will not measure the force transmitted from the drivers to the supporting wheels, unless corrected for the initial condition of stress in the draw-bar. It is obvious, therefore, that if comparisons are to be made between the power developed in the cylinders and that delivered

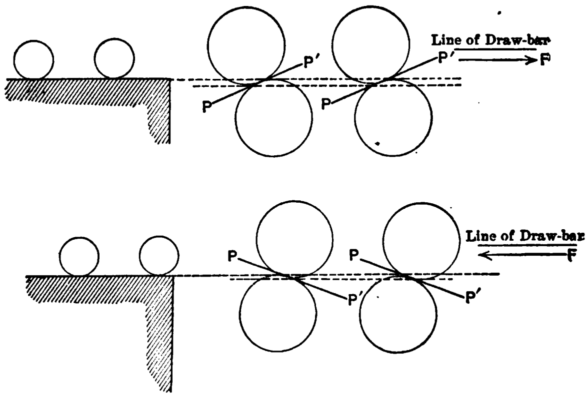


FIG. 204.

at the draw-bar the position of the drivers upon the supporting wheels must be neutral, or their position must be so accurately known that correction may be applied.

The significance of all this was appreciated from the first, though the difficulties in the problem were not anticipated. When the locomotive had been installed upon the plant it had been carefully adjusted to its position while cold. The failure to get consistent records at the draw-bar suggested that the spacing of the driving-axles might change sufficiently with the increased temperature of the frame resulting from working conditions to account for the difficulties encountered. Consequently, when the work was next undertaken (September, 1895), the wheels were lined up when the engine and boiler of the locomotive were heated to working temperature. Moreover the effect of machine stresses was also taken into account, all measurements being repeated with steam in the cylinders, first with

the reverse-lever in forward motion, and afterward with the reverse-lever in backward motion, the supporting wheels being prevented from turning by means of the friction-brakes. This process was repeated for various positions of the crank. The completion of this programme of preliminary work failed to disclose any gross errors in the position which the engine had occupied in the previous tests, and trial runs to determine values for engine friction proved no more satisfactory than those which had resulted from previous attempts. Meanwhile, the locomotive was being occupied with work not depending upon accurate records of draw-bar stress.

Up to this time the traction dynamometer, an expensive machine selected especially because of the reputation of its type for accuracy, had been accepted as a standard of measure No one had doubted its accuracy. But the experiences already recounted, in combination with other events at the laboratory, required that some effort be made to check its indications. This was finally done by moving an Olsen testing-machine to the locomotive laboratory, by disconnecting the dynamometer from the locomotive, and by mounting the hydraulic head of the dynamometer, with all attached parts, upon the testing-machine in such a manner that a load could be imposed upon it in the testing-machine through the same spindle which had previously connected with the locomotive draw-bar. By these means it was possible to apply to the dynamometer any desired load as measured by the beam of the testing-machine, and to determine the indication which the dynamometer scale gave in response to these loads. The degree of accuracy attending the work, of course, was limited to the accuracy of the Olsen machine, but this proved quite sufficient for the purpose. The results of comparison proved that the indications of the dynamometer were too high and singularly irregular.

These results having been confirmed by a representative of the manufacturer, the whole machine was shipped to the factory for correction. It was there recalibrated, and in February, 1896, was restored to its place in the laboratory. Soon after attempts were again made to secure a measurement of the machine friction of the locomotive, but the results, while never showing negative friction and while much more consistent than those previously obtained, were still too discordant to be entirely satisfactory, whereupon nothing remained but to again give attention to the location of the wheels of the locomotive. That this might be more carefully investigated than on previous occasions a telltale was arranged to connect between the locomotive

frame at a point midway between the two drivers and the wall of the laboratory (Fig. 205). It consisted of a small pointer, suitably mounted, and so proportioned as to magnify the actual movement of the locomotive approximately ten times. The end of the pointer passed over a stationary scale graduated from an arbitrary zero by divisions of equal value, each of which represented an actual movement of the engine-frame of an .0543 of inch. The value of this arrangement was to be found in the fact that any position of the pointer indicated at all times a certain relative position of drivers and supporting wheels. By its aid the problem of locating the drivers upon the supporting wheels resolved itself into the finding of a "position correction" corresponding to the different positions of this pointer.

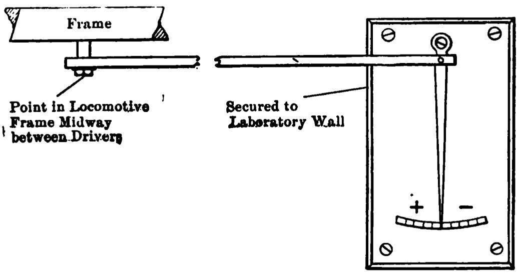


FIG. 205.

As it was found that correct values for draw-bar stress could not be read directly from the dynamometer, an attempt was made to determine the amount of the position correction by direct weighings. The method of procedure was to move the locomotive by means of a jack-screw, until the pointer stood at zero, and with an initial load on the dynamometer to balance the scale-beam. Next, without changing the initial load, the locomotive was moved by means of the jack until the pointer stood at each of the several different division marks on the forward scale, and thence back to zero, the dynamometer being balanced and readings taken for each position both forward and back. It was expected that when the dynamometer readings thus obtained were plotted, they would give a line which would determine the correction to be applied to the dynamometer readings when the engine was working in any of the given divisions experimented upon. The results, however, did not justify the pains, since the values obtained showed great irregularities, and, while the experiment was conducted with great care, was carried out with many variations,

and was many times repeated, it did not suffice to give a satisfactory position correction. he conclusion reached was to the effect that minor forces were brought into play whenever the engine was moved, and that the presence of these prevented the machine from behaving in the same way in every different position. It was thought, however, that these would disappear if similar tests could be made when the engine was in motion. To accomplish this involved the following program:

With the locomotive in any given position and running ahead, the pull registered by the dynamometer will be

$$D_F = M_F - F + x, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

where M_F = pull corresponding to the M.E.P. in the cylinders, engine running ahead;

F = pull equivalent to the machine friction;

x = pull due to engine position, which is the correction sought.

With the locomotive in the same position, running aback under the same conditions of speed, cut-off, and steam pressure as when running ahead, it may be assumed that the machine friction will be the same, and consequently that the draw-bar pull will be

$$D_B = M_B - F - x, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

where M_B = pull corresponding to the M.E.P. in the cylinders, engine running back;

F and x have the same values as above.

Solving these two equations for x, we have

$$x = \frac{(M_B - D_B) - (M_F - D_F)}{2}. \quad . \quad . \quad . \quad . \quad . \quad (20)$$

It will be seen that equation 20 makes the value of x, which is the correction for position, depend upon M and D. Numerical values for the former are obtained directly from the indicator-cards, and for the latter by direct observation from the traction dynamometer. The only assumption involved by the equation is that the friction of the engine is the same, when the same conditions of speed, cut-off, and initial pressure in the cylinders are maintained, whether the engine is running ahead or aback. The value of x, if considered positive when

the engine is running ahead, must be considered negative when the e gine is running aback.

In the application of this program a series of position tests were made in March, 1896, with results which were too much at variance one with another to be accepted as satisfactory. A month later it was determined to repeat the work with all of the friction-brakes removed from the axles of the supporting wheels, so that the power developed should approach closely the friction-load, whereupon it was found that with the engine running ahead there was compression in the draw-bar. An investigation of the cause of this unexpected result revealed the fact that the friction of the front trucks on the rails was too great to allow a free movement of the locomotive forward and back. Previous to this time but little attention had been given to the trucks, the belief being that the vibration of the engine in motion would be sufficient to overcome any resistance which they might offer.

A careful investigation at this time, however, revealed the fact that the rails which were mounted on wood had yielded under the pressure and vibration of the truck, becoming slightly deformed under each wheel. This fact having been determined, the truck was removed, the truck wheels, which had been cast in a contracting chill, were ground truly round, the wooden support for the rails was removed, and the work rebuilt in masonry and capped with metal. The rails were planed on their upper surface and drawfiled. The effect of these improvements was marked, and the results obtained from position tests which followed, when compared by means of equation 20, gave evidence for the first time of a definite law.

So much improvement had, in fact, been effected by bettering the mounting of the truck, that it was finally decided to entirely eliminate friction from this source by its entire removal. That this might be accomplished, the front of the engine was suspended from the roof of the laboratory, the construction of the building being such as readily to admit of such a plan (Fig. 206). Two rods, 17 feet in length, connected the front bolster with their point of support. With the engine at zero position upon the supporting wheels, these rods were made to stand vertically. Thus supported, the front of the engine was as free to move in any direction as a pendulum of equal weight.

With the engine thus suspended, tests were run to determine the indicated power and the tractive power for each position of the indicator (Fig. 205). By the manipulation of the turn-buckle connecting

between the draw-bar and dynamometer, the pointer of the telltale was allowed to occupy in succession the several positions along its scale, and in each position observations were made in triplicate. This was done with the engine running both ahead and aback. The reverse-lever was in all cases in its extreme position. The speed was as nearly as possible the same for each test, and the load was only that due

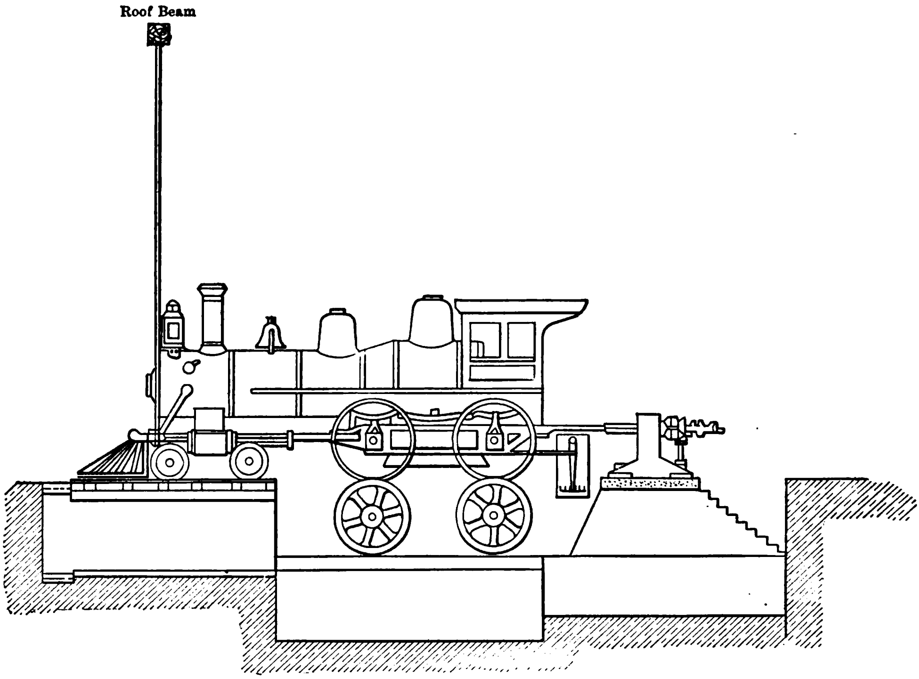


FIG. 206.—Method Used to Determine Position of Engine.

to the friction of the supporting-wheel axles. A summarized statement of the results of these tests appears as Table LXVIII., and the value of the position correction, as obtained by plotting these values, is shown by Fig. 207. The straight line of this diagram represents the average value of the readings given in Table LXVIII. The equation for this line is

$$x = -199 + 44\,y, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

where x = pull due to engine position (+ if tension, − if compression);
y = position of locomotive as shown by pointer (+ if forward, − if back of the arbitrary zero).

The point at which the mounting mechanism has no effect on the draw-bar is given as +4.52.

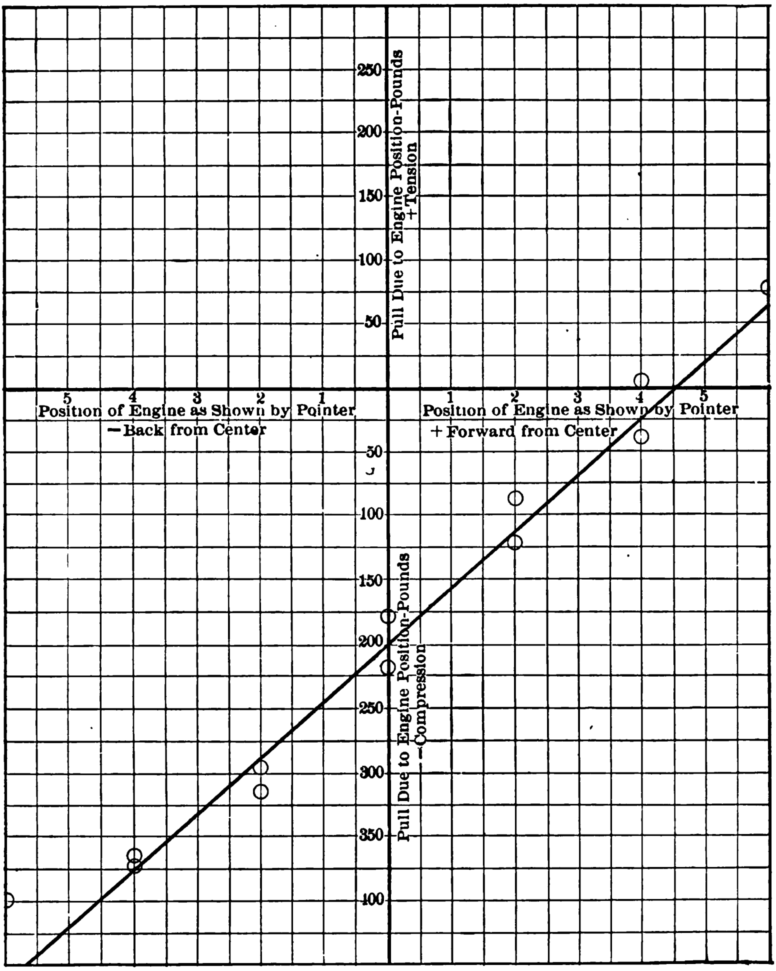


FIG. 207.—Draw-bar Pull Due to Engine Position.

When the locomotive is in the position corresponding with the zero position of the pointer, which, it will be remembered, was arbitrarily located, y will equal zero and

$$x = -199.$$

TABLE LXVIII.

INDICATED AND TRACTIVE POWER FOR DIFFERENT POSITIONS OF THE DRIVERS ON SUPPORTING WHEELS.

Position of Engine shown by Pointer.	Engine Running Ahead. Steam Pressure under Throttle, Lbs.	Speed, Revolutions per Minute.	Pull Corresponding to M.E.P. in the Cylinders, Lbs.	Pull shown by Dynamometer, Lbs.	Pull Corresponding to M.E.P. in the Cylinders Minus Pull shown by Dynamometer, Lbs.	Engine Running Aback. Steam Pressure under Throttle, Lbs.	Speed, Revolutions per Minute.	Pull Corresponding to M.E.P. in the Cylinders, Lbs.	Pull shown by Dynamometer, Lbs.	Pull Corresponding to M.E.P. in the Cylinders Minus Pull shown by Dynamometer, Lbs.	Pull Due to Engine Position, Lbs. (½ Algebraic Sum of Columns VI and VIII) + Ind. Tension, − Ind. Compression.
I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.
0	29	54	2552	2245	307	25	53	2151	2203	−52	−179
+2	27	55	2405	2168	237	25	52	2146	2154	−8	−123
+4	25	61	2155	2016	139	26	51	2225	2162	63	− 38
+6	29	48	2544	2532	12	27	49	2315	2149	166	+ 77
+4	27	53	2300	2210	90	27	48	2264	2168	96	+ 3
+2	28	51	2397	2209	188	24	58	1925	1913	12	− 88
0	27	50	2368	2097	271	22	59	1848	1932	−84	−178
−2	26	52	2276	1903	373	25	52	2129	2345	−216	−295
−4	24	56	2097	1657	440	25	52	2183	2490	−307	−374
−6	23	53	2071	1595	476	25	51	2193	2516	−323	−400
−4	23	56	2042	1565	423	26	50	2249	2560	−311	−367
−2	23	57	1968	1582	386	25	49	2212	2456	−244	−315
0	23	56	2051	1765	286	23	49	2295	2448	−153	−220

That is, under these conditions, 199 pounds must be subtracted from the observed reading of the dynamometer; corrections for other positions may, of course, be read directly from the diagram (Fig. 207), or found by means of equation 21.

It will be evident, also, that having the data which appears in Table LXVIII., the value of the correction for any position of the locomotive may easily be calculated by means of equation 20. The values given are: In Col. IV, M_F; in Col. V, D_F; in Col. IX, M_B; and in Col. X, D_B of this equation. The application of this equation to the data of any given position will give a result consistent with that which appears in the same line under Col. XII.

With reference to the accuracy of the foregoing conclusions, it should be noted that data was obtained for all positions of the locomotive from 1.5 to −.25, a range which extends far beyond the limits of Table LXVIII. and Fig. 207. The extreme values, while of use in

determining the slope of the plotted line, are omitted as unnecessary to the present discussion.

Finally, with reference to the long and laborious process which has been described, the fact should be emphasized that most of the difficulties encountered were of a kind which, in the light of past experience, should have been easily avoided. It would be wholly wrong to conclude that a locomotive testing plant, constructed along the lines developed at Purdue, is defective as an instrument for measuring the tractive power of a locomotive, or that accuracy of observed data is obtained at the expense of unusual effort. It should be clear that defects in the rating of the dynamometer, which for so long a time proved so perplexing, should not be charged against the principle underlying the plant. The importance of one rule of practice, however, is emphasized by the experience in question. It is to the effect that, for very accurate work, too much reliance should not be placed upon vibration as a means of overcoming truck friction. The rail surface should be in perfect form. If the same locomotive is to be operated on a plant for a considerable time, it will be necessary frequently to inspect the rails, to make sure that corrosion or side movement of wheels has not formed incipient grooves under the tread of each wheel. If care is given this matter, and if the drivers are made plumb over the supporting wheels by careful measurement, the record of the dynamometer will be sufficiently accurate for every practical purpose, even though the stresses to be measured are relatively light.

169. **Friction Tests and their Results.**—Having obtained a dynamometer which could be depended upon, and having determined the engine position with such care as to permit the measurement of draw-bar stresses with a high degree of accuracy, a series of tests to determine machine friction were undertaken. In proceeding with the work, the locomotive having been warmed by preliminary running, was brought under conditions for which information was desired. Upon signal, indicator-cards were taken, the draw-bar pull ascertained, and all running conditions observed. All observations were taken simultaneously, and were three times repeated at intervals of four minutes, after which other conditions of running were sought and another test was entered upon. Tables LXIX. and LXX. which in effect constitute a single table, give in each line the average of the three observations making up one test and the calculated values obtained therefrom. The running conditions are set forth by Cols. 2 to 7 the power developed in the cylinders and at the draw-bar by

TABLE LXIX.

RESULTS OF FRICTION TESTS.

Number.	Speed.		Cut-off.		Pressure		M.E.P.				
	Miles per Hour.	R.P.M.	Notch Forward from Center.	Per Cent of Stroke.	Boiler.	Dry-pipe.	Right.		Left.		Average.
							H.E.	C.E.	H.E.	C.E.	
1	2	3	4	5	6	7	8	9	10	11	12
1	14.35	77.22	1	25	132	128	42.87	51.77	46.95	47.82	47.35
2	15.04	80.94	1	25	128	124	37.30	44.98	42.17	44.98	42.36
3	25.16	135.4	1	25	129	124	34.05	42.67	38.98	39.64	38.83
4	25.70	138.3	1	25	130	127	33.51	40.71	35.75	39.24	37.30
5	35.64	191.8	1	25	132	126	30.64	36.86	34.33	35.91	34.43
6	35.47	190.9	1	25	128	125	25.98	33.33	30.71	34.30	31.00
7	46.68	251.2	1	25	132	129	23.17	28.30	27.69	30.00	27.29
8	56.83	305.8	1	25	131	127	18.42	24.34	24.49	26.09	23.33
9	14.29	76.89	2	35	129	125	57.16	63.74	61.49	61.85	61.06
10	24.94	134.2	2	35	124	119	46.84	53.03	51.62	52.7[illegible]	51.07
11	35.96	193.5	2	35	126	121	38.79	45.23	44.66	46.78	43.86
12	46.48	250.6	2	35	125	121	32.04	37.61	37.57	38.83	36.51
13	58.06	312.4	2	35	123	122	25.04	29.75	30.64	31.74	29.29
14	25.48	137.1	3	45	131	128	61.78	66.82	69.65	68.68	66.73
15	36.70	197.5	3	45	125	118	49.85	54.68	56.10	56.50	54.28
16	14.38	77.40	9	80	135	81	65.74	68.26	70.44	72.13	69.14
17	14.74	79.30	9	80	132	73	60.06	62.49	62.17	63.35	62.02
18	35.61	191.6	2	35	148	104	31.23	37.26	37.05	38.48	36.00
19	35.66	191.9	2	35	126	100	30.66	35.75	35.35	35.93	34.42
20	35.88	193.1	2	35	95	95	27.77	31.80	31.56	33.61	31.18
21	35.59	191.5	2	35	76	76	21.35	24.99	22.91	25.10	23.59
22	24.16	130.0	1	25	127	124	32.86	44.16	40.00	41.64	39.67
23	25.09	135.0	1	25	129	125	33.00	45.80	41.30	43.24	40.83
24	25.18	135.5	1	25	124	122	32.77	44.57	40.05	43.23	40.15
25	25.46	137.0	1	25	126	123	33.61	45.45	41.02	43.40	40.87
26	25.27	136.0	1	25	131	129	34.69	43.79	41.24	41.46	40.34
27	25.18	135.5	1	25	129	127	34.47	41.22	39.73	39.78	38.80
28	25.46	137.0	1	25	132	130	35.27	41.57	41.14	41.35	39.83
29	26.48	142.5	1	25	130	124	30.78	39.34	36.37	36.59	35.77

Cols. 9 to 18, and the friction loss as expressed in various terms by Cols. 19 to 22. The dynamometer pull equivalent to the indicated horse-power (Col. 18), depends upon the relation of cylinder dimensions to diameter of drivers. It assumes no loss of power in transmission from the cylinder to draw-bar and is expressed by the equation

lbs. M.E.P. in each cylinder = .009139 times lbs. pull at the draw-bar.

TABLE LXX.

RESULTS OF FRICTION TESTS—(*Continued*).

Number.	I.H.P.	Dynamometer.					Friction in Terms of			
		Actual Reading.	Correction (to be added).	Dynamometer Reading (corrected).	D.H.P.	Dynamometer Pull Equivalent to I.H.P.	I.H.P.	Per Cent of I.H.P.	Pounds Dynamometer Pull.	Pounds M.E.P.
1	13	14	15	16	17	18	19	20	21	22
1	198.1	4,392	166	4,558	174.5	5,177	23.6	11.95	619	5.66
2	185.7	3,936	133	4,069	163.2	4,630	22.5	12.12	561	5.13
3	285.0	3,472	168	3,640	244.4	4,245	40.6	14.25	605	5.53
4	285.6	3,449	133	3,582	245.4	4,169	40.2	13.07	587	5.40
5	357.8	3,015	169	3,182	302.4	3,764	55.4	15.46	582	5.33
6	321.3	2,681	141	2,822	267.0	3,396	54.3	16.90	574	5.24
7	371.4	2,292	194	2,486	309.5	2,983	61.9	16.66	497	4.49
8	396.1	1,799	205	2,004	303.9	2,614	92.2	23.31	610	5.57
9	254.4	6,045	141	6,186	235.6	6,678	18.8	7.38	492	4.50
10	371.4	4,965	151	5,116	340.1	5,585	31.3	8.47	469	4.28
11	460.0	4,188	155	4,343	416.6	4,796	43.4	9.43	453	4.14
12	495.6	3,402	184	3,586	445.3	3,992	50.3	10.12	406	3.71
13	497.1	2,524	184	2,708	419.1	3,212	78.0	15.65	504	4.61
14	495.9	6,768	130	6,898	468.8	7,297	27.1	5.49	400	3.66
15	582.0	5,424	134	5,558	544.4	5,939	37.6	7.31	381	3.48
16	290.1	7,284	136	7,420	284.6	7,562	5.5	1.89	142	1.30
17	266.6	6,416	141	6,557	257.7	6,784	8.9	3.33	227	2.08
18	373.9	3,349	167	3,516	333.9	3,936	40.0	10.70	420	3.84
19	358.0	3,193	135	3,328	316.6	3,763	41.4	11.62	435	3.98
20	327.5	2,812	128	2,940	281.3	3,423	46.2	14.11	483	4.41
21	245.1	1,978	130	2,108	199.9	2,584	45.2	18.42	476	4.35
22	285.4	3,732	156	3,888	250.5	4,430	34.9	12.23	542	4.95
23	298.5	3,617	130	3,747	250.7	4,462	47.8	16.01	715	6.53
24	294.8	3,592	149	3,741	251.2	4,390	43.6	14.79	649	5.93
25	303.2	3,580	144	3,724	252.8	4,466	50.4	16.62	742	6.78
26	296.9	3,662	140	3,802	256.2	4,405	40.7	13.71	603	5.51
27	284.9	3,472	137	3,609	242.4	4,242	42.5	14.91	633	5.78
28	295.7	3,650	133	3,783	256.8	4,356	38.9	13.12	573	5.24
29	276.1	3,225	130	3,355	236.9	3,910	39.2	14.19	555	5.07

170. A Comparison of Results.—The factors, upon which the loss of power by friction in the machinery of a locomotive depends, concern chiefly the lubrication of the moving parts. If a viscous oil is used in any or all of the journals it should be expected that the frictional loss will be greater than when a more limpid oil is used. Again, with a given lubricant in service, the coefficient of friction

is affected by the temperature of the lubricant and of the metallic parts surrounding. Within limits which are rather wide the friction diminishes as the temperature of the lubricant is increased. In view of these facts, therefore, attempts to measure frictional loss should not be expected to give results which are free from inconsistencies, since it is obviously impracticable to control the temperature of all the journals of a locomotive, or even the precise rate at which lubricant is supplied them. The most that can be attempted is to secure results under a sufficient number of different working conditions to fairly represent the ordinary running conditions of the locomotive. Minor inconsistencies between individual results must be expected.

The preceding statements will naturally suggest the fact that the operation of a locomotive, when first started, is attended by greater frictional loss than after its parts have become well warmed by running. The truth of this statement was demonstrated by actual test. After being at rest for a period of twenty-four hours the Purdue locomotive was started and brought to a speed of 25 miles an hour, with the reverse-lever in the first notch forward of center and the throttle fully open. As soon as possible after starting, observations necessary to a determination of the frictional loss were made, and were thereafter repeated at five-minute intervals for a period of thirty minutes, after which the interval between observations was increased to ten minutes, and still later it was further increased to twenty minutes. At the end of the first ten minutes the friction, as expressed in pounds M.E.P., was $6\frac{1}{2}$; after twenty minutes it had fallen to 5.29, after which the friction fluctuated by small amounts both above and below this value throughout the remainder of the run, which was continued for 110 minutes.

Referring to Tables LXIX. and LXX. it will be seen that all tests of the first series (Tests 1 to 8) were made with the same cut-off, a constant boiler pressure, and a fully open throttle. Tests in duplicate at speeds approximating 15, 25, and 35 miles respectively, together with single tests at speeds of 47 and 57 miles, are included in this series. The duplicates represent the work of different days, the purpose being to show whether the conditions of lubrication, and, consequently, of engine friction, vary materially from day to day. Comparing the results derived from these duplicates, one with another, by means of values given in Cols. 19 to 22, it will appear that differences are slight.

A study of the whole series justifies the conclusion that frictional

resistance, as shown by stress in the draw-bar, is approximately constant for all speeds, being equivalent to a little more than 5 pounds mean effective pressure, or about 600 pounds pull at the draw-bar. As measured in horse-power the frictional loss under these conditions increases with each increase in speed.

The second series (Tests 9 to 13) were run under conditions sim lar to those of the preceding group, except that the reverse-lever was in the second notch from the center forward, giving a cut-off of 35 per cent. In this case, also, the frictional loss, as measured by M.E.P. or by draw-bar stress, is practically constant for all speeds, but its value is numerically less than when the cut-off is 25 per cent.

The third series (Tests 14 and 15) show the same progress in relationship between cut-off and friction as noted for the previous group, the cut-off in this case being very nearly half-stroke, and the friction being no more than two-thirds that which attends operation under a cut-off of 25 per cent.

The fourth group of values (Tests 16 and 17) was obtained with the reverse-lever in the ninth notch from the center, giving a cut-off of 80 per cent, the throttle but partially open. All conditions of running were the same for both tests, except that in Test 16 the throttle-opening was such as to give a dry-pipe pressure of 81 pounds; and, in Test 17, a dry-pipe pressure of 73 pounds. The tests are very nearly duplicates of each other, so far as running conditions are concerned. The resultant friction is lower than for any condition previously noted.

The series embraced by Tests 18 to 21 also represent tests run under different degrees of throttling. In this case the reverse-lever was in the second notch, making the cut-off 35 per cent. The values for frictional resistance (Col. 22) are nearly constant, though they are higher than those obtained from either of the tests of the preceding group.

Tests 22 to 29, unlike those which precede them, represent data which were drawn from formal efficiency tests. In this case each value is the average of from fifteen to thirty different observations. When compared with the values of preceding groups, representing similar conditions of running, they will show the degree of coincidence in the results secured from special tests and those which are derived from the more formal operation of the plant.

171. Conclusions.—From a comparative study of the results presented in Tables LXIX. and LXX. it appears that when the cut-off remains unchanged, the force necessary to overcome the frictional

resistance of the machinery of a locomotive is independent of speed. This force may be expressed in terms of mean effective pressure or of pull at the draw-bar.

The experimental results make it evident, also, that the force necessary to overcome the friction of the machinery diminishes as the pressure throughout the stroke becomes more uniform. As expressed in terms of the data it diminishes with increase of cut-off. This is true for a wide-open throttle, and it continues to be true when the cut-offs are made so long that the throttle must be partially closed to avoid exhausting the boiler. With a given dry-pipe pressure the force necessary to overcome the frictional resistance is evidently least when the reverse-lever occupies an extreme position upon the quadrant. All of these facts, as they apply to the locomotive experimented upon, may be summarized as follows:

When the reverse-lever is in first notch (cut-off, 25% of stroke), wide-open throttle,

the frictional resistance = 5.29 pounds M.E.P.
= 579 " at the draw-bar.

When the reverse-lever is in second notch (cut-off, 35% of stroke), wide-open throttle,

the frictional resistance = 4.18 pounds M.E.P.
= 458 " at the draw-bar.

When the reverse-lever is in third notch (cut-off, 45% of stroke), wide-open throttle,

the frictional resistance = 3.57 pounds M.E.P.
= 391 " at the draw-bar.

When the reverse-lever is in ninth notch (cut-off, 80% of stroke), partially open throttle,

the frictional resistance = 1.69 pounds M.E.P.
= 185 " at the draw-bar.

With the force necessary to overcome the frictional resistance of the machinery constant, as set forth above, it is evident that the power absorbed in friction will be proportional to the speed. The facts with reference to the horse-power equivalent of the machine friction for the engine tested may be defined as follows:

When the reverse-lever is in first notch (cut-off, 25% of stroke), wide-open throttle,

machine friction expressed in horse-power = 1.39 × speed expressed in miles per hour.

When the reverse-lever is in second notch (cut-off, 35% of stroke), wide-open throttle,

machine friction expressed in horse-power = 1.24 × speed expressed in miles per hour.

When the reverse-lever is in third notch (cut-off, 45% of stroke), wide-open throttle,

machine friction expressed in horse-power = 1.04 × speed expressed in miles per hour.

The preceding statement has been made the basis from which to determine a correction for the draw-bar stress, applying to the data of all tests run prior to 1896, which are given in Chapter IV. For these early tests the draw-bar stress repo ted is not an observed value, but one which has been thus deduced.

V. LOCOMOTIVE PERFORMANCE.

CHAPTER XX.

THE EFFECT OF THROTTLING.

172. **Throttling.**—A locomotive which is being operated under a partially closed throttle-valve is, in the language of the road, being operated under the throttle; its output of power is then controlled by the position of the throttle-valve. For operation on the road it is necessary, to avoid exhausting the boiler, that some general relation be observed between the position of the reverse-lever and that of the throttle. As the cut-off is reduced by hooking up the reverse-lever, the throttle-opening may be increased. This fact in the earlier days led to many discussions among locomotive engineers as to whether it were better to run a locomotive with a long cut-off and a relatively small throttle-opening or to reverse these conditions. As to the merits of the discussion, it should be said that there was a time when the machinery of locomotives was so light as to lend color to the belief that the locomotive ran better, and in return for a smaller consumption of fuel, when the pressure of steam admitted to the cylinders was much below that of the boiler; but in recent years locomotives have become better designed, and practice has tended steadily toward the wide-open throttle. In so doing it has given a true response to well-known thermodynamic principles.

While correct theory points to the desirability of admitting steam to the cylinders at as high a pressure as practicable, the losses in cylinder efficiency, resulting from a 20 or 30 per cent reduction of pressure by throttling, are not large. This is due to the fact that in wiredrawing past the throttle the quality of the steam supplied the cylinders is improved; moist steam is dried, and steam which initially is dry or nearly so, may be superheated. Because of this action the loss resulting from the drop in pressure of the steam is in part neutralized by the rise in its quality.

173. **The Tests** designed to disclose the effect of different degrees of throttling upon the economic performance of the engine are of two groups. The first group of nineteen tests represents the first work undertaken upon the Purdue testing-plant. These tests were run at a time when the experimental character of the plant itself required that the locomotive be worked at less than its maximum power. The second group consists of three tests which were run at a much later

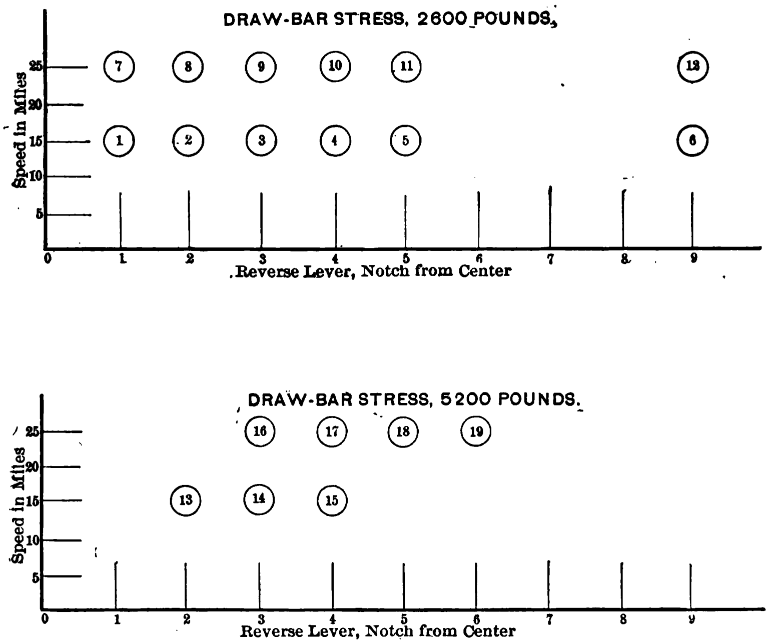


FIG. 208.—Conditions Governing Tests under the Throttle.

date, the results of which are recorded in Chapter IV.* Dealing first with the tests of the first group, the conditions of speed and cut-off employed are defined by Fig. 208. It will be seen by this figure that the tests were arranged in two series, for which the draw-bar stress was constant at 2600 pounds and 5200 pounds respectively.

* The first group of nineteen tests were run in 1892–93, and were described in detail in a paper entitled "Tests of the Locomotive at the Laboratory of Purdue University," presented to the American Society of Mechanical Engineers, July, 1893. The full exhibit of data from these tests is omitted from the record of Chapter IV.

The constant draw-bar stress for each series was secured for the several speeds and cut-offs chosen by the manipulation of the throttle.

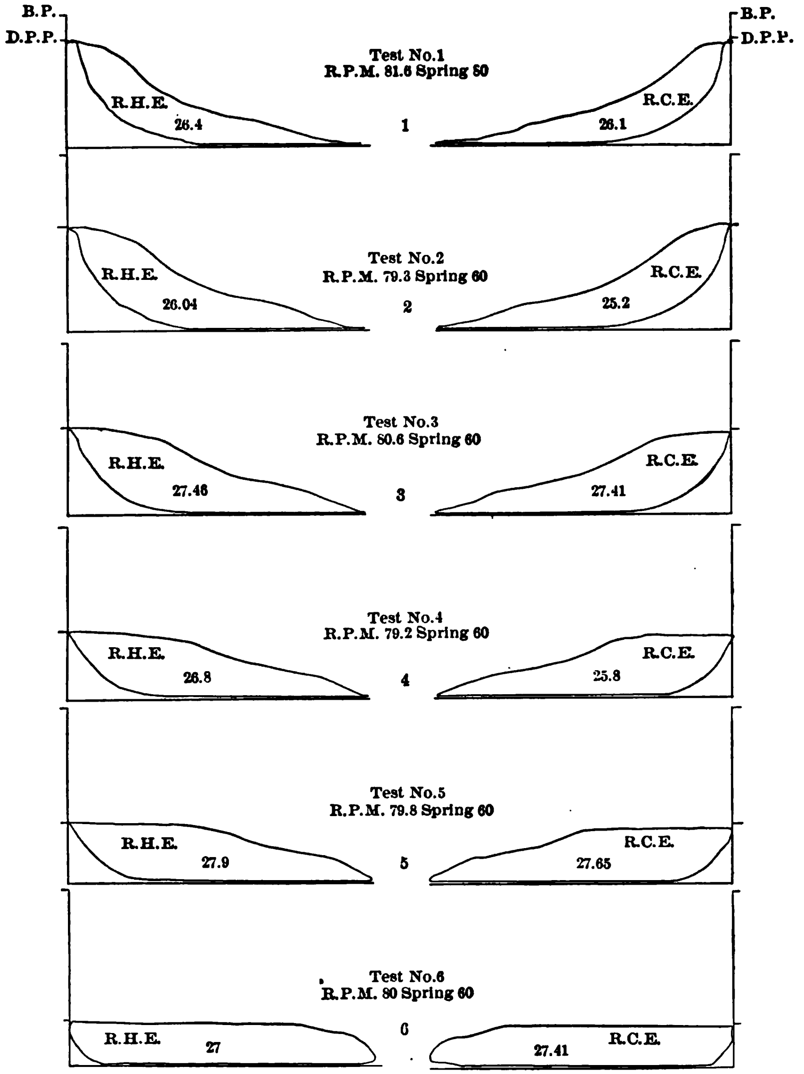


FIG. 209.

174. Indicator-cards.—Typical indicator-cards, resulting from the process described, are given as Figs. 209, 210, 211, and 212. An inspection of the cards for any series will show that the mean effective

pressure is practically constant, a condition growing out of the requirement with reference to a constant draw-bar stress. It will be seen,

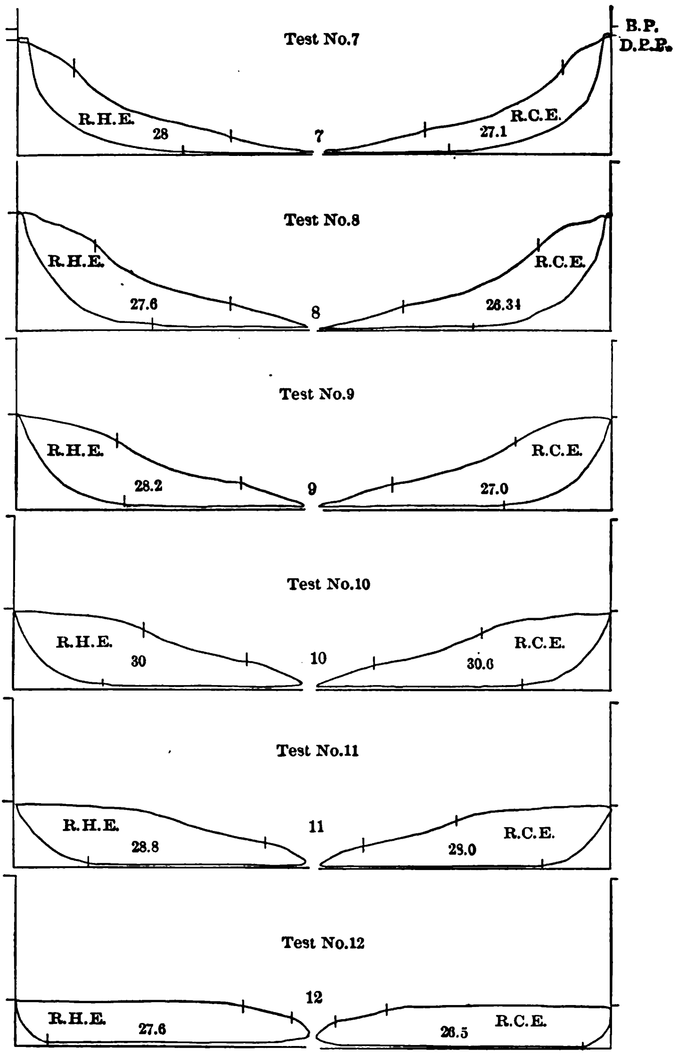


FIG. 210.

also, that the degree of throttling is slight for the shorter cut-offs, and that it increases step by step as the cut-off is lengthened, until the initial pressure is but a small fraction of that of the boiler. The

whole group of tests includes four series, for each of which the element of progressive throttling appears thus: Under a draw-bar stress of 2600 pounds, a series at 15 miles an hour and another at 25 miles an hour; under a draw-bar stress of 5200 pounds, a series at 15 miles an hour and another at 25 miles an hour.

175. **Numerical Results** which concern the present purpose are presented as Tables LXXI. and LXXII. The test numbers of these

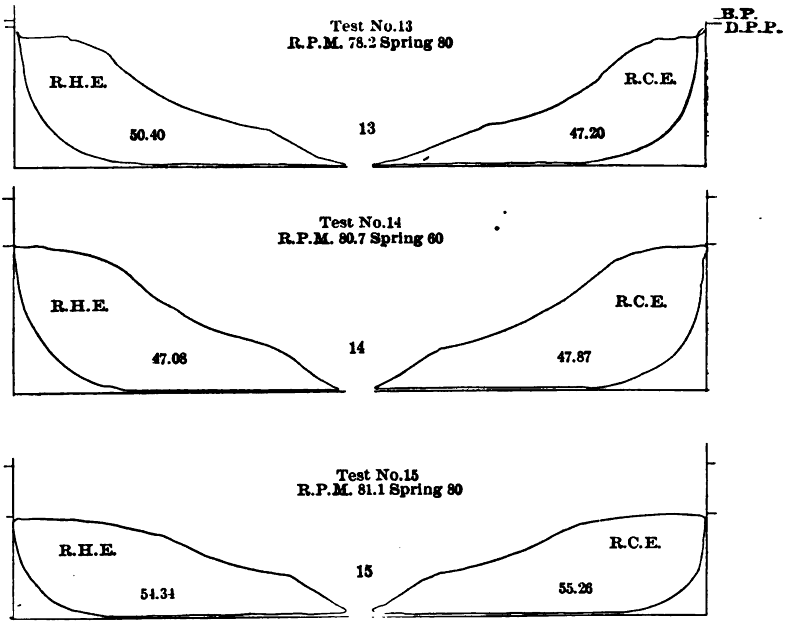


Fig. 211.

tables agree with the numbers of Fig. 208, and also with the numbers assigned to the indicator-cards in Figs. 209 to 212. The boiler pressure, Col. 7, is practically constant for all tests, and the difference between this and the dry-pipe pressure (Col. 8) may be accepted as a measure of the degree of throttling in each case. By Col. 9 it will be seen that when the power developed is light, the steam in the dome of the boiler is so dry that it becomes considerably superheated in passing the throttle, the degree of superheating increasing as the difference in pressure upon the two sides of the throttle is increased. As the power developed becomes greater, the quality of the boiler steam falls and the superheating effect necessarily diminishes. The failure of the steam to superheat during the tests of highest power, however,

is in part to be accounted for in the lesser degree of throttling which occurs in these tests.

The steam consumption per horse-power per hour (Col. 11) shows the extent to which the economy of the engine declines with each increase of the throttling action.

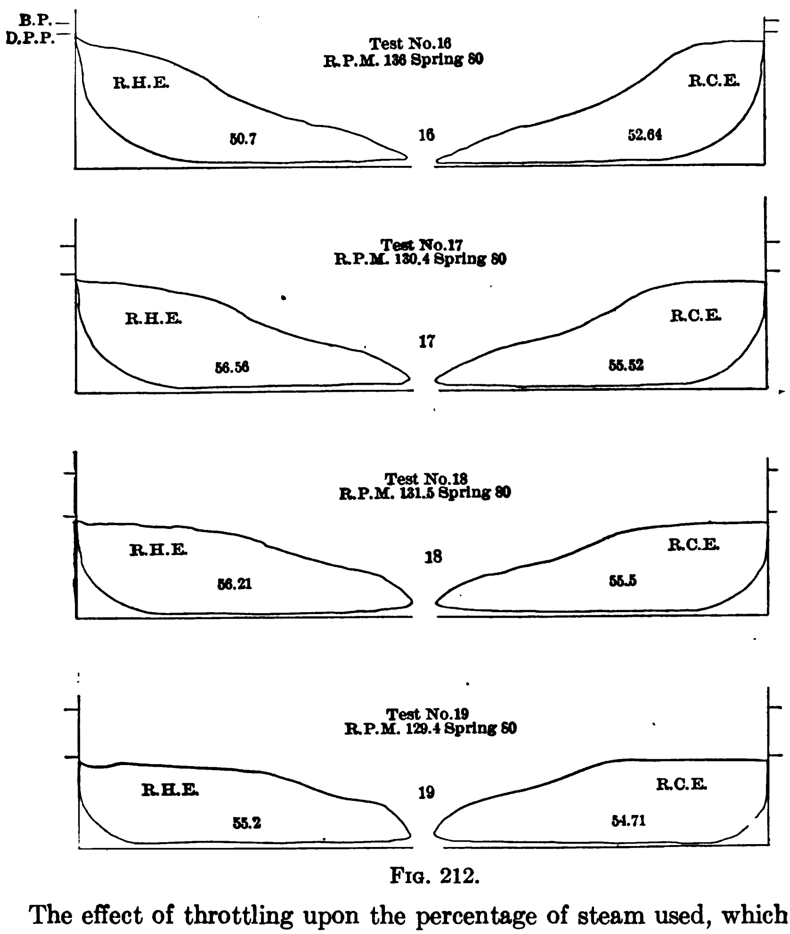


FIG. 212.

The effect of throttling upon the percentage of steam used, which is shown by the indicator, is given in Cols. 12 and 13, and in this connection it is of interest to observe that the percentage of steam, which is accounted for by the indicator, is increased by any increase in the degree of throttling or in the period of admission. Under the extreme conditions covered by the tests, as much as 95 per cent of all the steam supplied the cylinders appears in the record of the indicator.

The reëvaporation or condensation during a revolution (Cols. 15 and 16) is chiefly affected by changes in cut-off. As the cut-off is lengthened, the period of expansion is diminished, and the range of temperature to which the walls are exposed becomes reduced, so that the whole process of heat interchange diminishes. Moreover, since these changes serve to shorten the expansion curve, the period to which the observations apply is shortened, all of which serve to explain the gradual diminution in the amount of reëvaporation and the appearance of condensation in its place.

TABLE LXXI.

RESULTS FROM THROTTLING-TESTS.

	Number.	Speed.		Cut-off.		Boiler Pressure.	Dry-pipe Pressure.	Super-heat in Dry Pipe, Degrees F.	I.H.P.	Steam (by Tank) per I.H.P. per Hour, Lbs.
		Miles per Hour.	R.P.M.	Notch from Center.	Per Cent of Stroke.					
1	2	3	4	5	6	7	8	9	10	11
Series 1	1	15.2	81.60	1	18.6	128.9	104.4	0	128.4	30.24
	2	14.8	79.30	2	26.9	129.8	76.0	4	128.5	30.13
	3	15.1	80.59	3	36.2	126.8	62.0	11	128.1	31.28
	4	14.8	79.18	4	46.7	125.5	48.6	22	117.7	35.40
	5	14.9	79.80	5	55.5	128.7	43.0	24	124.3	36.67
	6	14.9	80.02	9	77.3	129.7	32.1	35	118.2	47.07
Series 2	7	23.8	127.37	1	18.6	128.4	117.8	0	212.3	28.97
	8	24.0	128.35	2	26.9	129.6	89.0	0	205.7	29.18
	9	24.0	128.47	3	36.2	130.3	70.0	8	209.9	30.33
	10	24.1	128.77	4	46.7	130.3	58.9	18	220.6	31.56
	11	23.9	128.14	5	55.5	127.7	48.0	22	205.3	37.06
	12	23.9	128.15	9	77.3	129.7	33.9	31	188.5	44.62
Series 3	13	14.6	78.20	2	26.9	130.0	126.1	0	222.9	26.77
	14	15.1	76.42	3	36.2	129.6	97.3	0	224.8	29.48
	15	15.2	81.13	4	46.7	130.4	89.2	0	244.5	30.04
Series 4	16	25.4	136.04	3	36.2	129.3	118.0	0	399.5	24.97
	17	24.4	130.40	4	46.7	130.2	105.3	0	412.2	27.57
	18	24.6	131.50	5	55.5	128.0	91.0	0	412.3	29.62
	19	24.2	129.37	9	77.3	123.0	79.3	0	392.5	32.08

176. Steam Consumption.—The weight of steam used by the locomotive per I.H.P. hour, when developing a constant amount of power under different degrees of throttling, is shown by Fig. 213, the curves of which are plotted from the values of Tables LXXI. and LXXII. A glance at this figure is sufficient to show that throttling effects an

TABLE LXXII.

RESULTS FROM THROTTLING-TESTS—(*Continued*).

	Number.	Steam (by Indicator) per I.H.P. per Hour, Approximate. Lbs.	Per Cent of Mixture in Cylinder Present as Steam at Cut-off.	Per Cent of Live Steam Condensed During Admission.	Reëvaporation per Revolution. Lbs.	Condensation per Revolution. Lbs.
1	2	12	13	14	15	16
Series 1	1	17.50	70	42	.0643	0
	2	19.76	78	30	.0617	0
	3	23.83	81	24	.0272	0
	4	28.93	85	18	.0049	0
	5	31.08	87	15	.0094	0
	6	42.8	92	13	0	.0193
Series 2	7	17.85	73	39	.0285	0
	8	20.72	79	29	.0081	0
	9	23.65	83	22	.0125	0
	10	26.2	86	17	.0069	0
	11	30.38	85	18	0	.0045
	12	42.22	95	9	0	.0193
Series 3	13	18.8	75	30	.0111	0
	14	20.42	77	26	.0034	0
	15	24.00	83	16	0	.0004
Series 4	16	20.67	84	20	.0068	0
	17	24.37	88	14	0	.0113
	18	26.09	89	12	0	.0011
	19	29.00	90	10	0	.0036

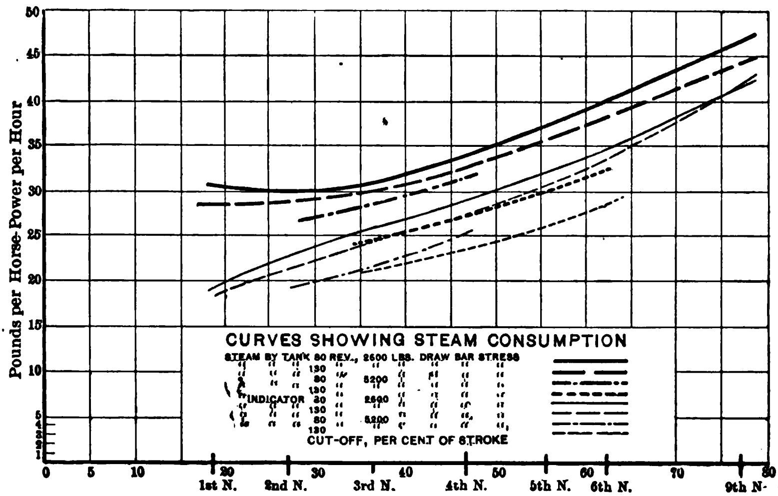


FIG. 213.

TABLE LXXIII.

RESULTS FROM THROTTLING-TESTS—(*Continued*).

	Number.	Indicated Horse-power.	Tractive Horse-power.	Friction of Engine in Horse-power.	Friction of Engine, Per Cent of Indicated Horse power	Dynamometer Work in Foot-tons per Pound of Steam by Tank.
1	2	17	18	19	20	21
Series 1	1	128.38	106.03	22.35	17.41	26.90
	2	128.47	103.05	17.43	14.47	28.07
	3	128.09	104.72	23.37	18.24	25.82
	4	117.75	102.89	14.86	12.62	24.38
	5	124.30	103.70	20.60	16.57	22.45
	6	118.21	103.96	14.25	12.05	18.49
Series 2	7	212.37	165.51	46.85	22.06	26.63
	8	205.70	166.79	38.91	18.92	27.50
	9	209.93	166.95	42.97	20.47	25.96
	10	220.61	167.44	53.17	24.10	23.58
	11	205.31	166.52	38.79	18.89	21.67
	12	188.51	166.52	21.97	11.65	19.64
Series 3	13	222.90	200.83	22.06	9.89	33.33
	14	224.85	209.28	15.56	6.92	31.52
	15	244.56	210.50	34.06	13.92	28.64
Series 4	16	399.48	353.90	44.09	11.08	33.53
	17	412.23	338.36	73.87	17.92	29.47
	18	412.28	341.19	71.09	16.90	27.60
	19	392.51	335.67	56.84	14.82	26.37

increase in the consumption of steam, the extent of the increase under the extreme conditions of the test amounting to more than 50 per cent. It is, however, noteworthy that, under very light power, a slight amount of throttling and a corresponding lengthening of cut-off affects the economic performance of the engine favorably. The extent of this favorable influence is shown in the upward turn of the lower end of the upper curve (Fig. 213). The results given may be accepted as a fair measure of the engine performance under different degrees of throttling while developing constant power. With reference to the question as to whether the engine should be run under the throttle or by the reverse-lever, they furnish direct and positive information.

From a strictly analytical point of view, however, it may be urged that all of the differences in effect which are shown should not be charged against the throttling action, since, as the throttling action

was varied, the cut-off also was changed. From this point of view, the curves of Fig. 213 may be regarded as showing differences in economy, resulting from differences in cut-off as well as from differences in the degree of throttling. The data, however, permit comparisons between tests of different series, which may be so selected that the cut-off of all will be the same. Thus, Tests 3, 9, 14, and 16 were all run with a cut-off of approximately 35 per cent, an efficient point for a wide-open throttle. Such a comparison is shown by Table LXXIV. which follows, from which it appears that the test having the least amount of throttling gives the most economical performance. Similar comparisons will in every case show similar results.

TABLE LXXIV.

EFFECT OF THROTTLING AT CONSTANT CUT-OFF, 35 PER CENT STROKE.

Test number	3	9	14	16
Miles per hour	15	25	15	25
Steam per I.H.P.	31.28	30.33	29.48	24.97
Dry-pipe pressure	62	70	97	118
Ratio of dry-pipe pressure to boiler pressure	.49	.54	.75	.91

Turning now to the second group of throttling-tests, results of which are reported in Chapter IV. (Nos. 12, 20, and 21), it will be found that the results confirm the conclusions already drawn. These tests were run at the same cut-off, but with varying degrees of throttling, a summary of results appearing as Table LXXV.

TABLE LXXV.

	Boiler Pressure.	Dry-pipe Pressure.	Ratio of Dry-pipe Pressure to Boiler Pressure.	Steam per I.H.P. per Hour.
35-2-A	131	121	.92	26.28
35-2-E	128	95	.74	27.92
35-2-F	155	93	.60	27.18

177. **Machine Friction.**—In the preceding discussion comparisons have been based upon the indicated horse-power. Reference to Table LXXIII. will make it apparent that, for the development of an equal amount of work, frictional losses are reduced as the cut-off

is lengthened and the throttle-opening diminished. One effect of throttling is to reduce engine friction. When, therefore, comparisons are based upon the work of the draw-bar, the results are somewhat more favorable to the practice of throttling. The modifying influence, however, of reduced friction is not sufficient to affect the genera conclusion already stated.

CHAPTER XXI.

EFFECT OF HIGH STEAM PRESSURES ON LOCOMOTIVE PERFORMANCE.

178. Power and Efficiency.—The power developed by a locomotive is a function of boiler pressure, steam distribution, diameter and stroke of piston, and of speed. The efficiency of a locomotive is a measure of the degree of perfection attending the development of the power; broadly stated, it is the ratio of the heat equivalent of the work done in the cylinders to the heat in the coal supplied to the fire-box. That engine is most efficient which, for each pound of coal burned, develops the largest amount of power in the cylinders. Anything which affects the efficiency, either of the boiler or of the engine of a locomotive, affects the efficiency of the locomotive as a whole.

The maximum power which can be developed by a given locomotive depends upon the power capacity of the several elements making up the complete machine. For example, any increase in its capacity for burning fuel will, other things being equal, increase the quantity of steam delivered from the boiler, giving an increased supply for the cylinders, and making it possible for them to develop a greater amount of power. Similarly, anything affecting the efficiency of the several transformations between the grate and the cylinder affects the maximum output of power. Thus, any increase in boiler efficiency will augment the amount of steam delivered for a given weight of coal burned, and steam thus obtained being available for the cylinders permits a higher rate of power. Again, anything which promotes the efficiency of the cylinder action will raise the maximum limit of power, since, with a constant supply of steam available, the most efficient cylinder action will result in the delivery of greatest power. There is, therefore, a double purpose gained in the adoption of any change in design or practice which improves the efficiency of any material part of the locomotive. If, in the presence of such a change, the output of power remains the same, then the increased efficiency

implies a saving in fuel, whereas, if the same quantities of fuel are consumed, then the increased efficiency implies a greater output of power.

It is the purpose of the present chapter to discuss briefly the relative advantage of different steam pressures employed in locomotive service. It is important in this connection to note that, as a problem of design, increase of pressure alone does not raise the limit of power. But if it can be shown that higher pressures improve the cylinder action, then their adoption will result in a saving of steam which, at the limit, may be made to appear as an increase of power. The experimental results which are quoted are those obtained from locomotive Schenectady No. 1. As the maximum pressure carried by this locomotive was but 140 pounds, the experiments deal with what is now to be regarded as a low range of pressure. The discussion, however, may take a broader view and the conclusions reached will, it is hoped, soon be confirmed by later experiments.

Steam pressure in locomotive service has been gradually increasing throughout the last two decades. In 1890, from 125 to 140 pounds were common, whereas, at this writing, pressures commonly range from 180 to 200 pounds, and a few locomotives are operating using 210 pounds. The term "high steam pressure" at this time may, therefore, properly apply to pressures above 180 pounds.

179. Thermal Advantages of High Steam Pressures.—Certain thermodynamic facts which underlie any discussion of the advantages of high steam pressure are presented as Table LXXVI.

Cols. I. and II. show the relation of pressure and temperature, while the rise of temperature for equal increments of pressure is found in Col. III. It is evident that the increased temperature for pressures above 180 pounds is not a very important factor, the rise from 180 to 250 being less than 30 degrees. Col. IV. shows how small an amount of heat is required to be added to steam of one pressure to convert it into steam of a higher pressure. The cost of high-pressure steam is but little more than that of a lower pressure. From theoretical considerations the performance of a perfect engine receiving steam at different pressures and exhausting against a back pressure of 13 pounds can be calculated. The results of such a calculation are presented in Cols. VI. and VII., an inspection of which will disclose the gain in economy resulting from each increment of pressure. The fact should be emphasized that the values given are ideal; they may be approached but never equaled by an actual engine. The values are important as showing that the gain for equal increments of pres-

TABLE LXXVI.

Gauge Pressure, Pounds.	Corresponding Temperature.	Increase in Temperature for each 25 Pounds Increase in Pressure.	Total Heat in One Pound of Steam above the Heat of Freezing Water, B.T.U.	Increase in Thermal Units in One Pound of Steam for each 25 Pounds Increase in Pressure.	Pounds of Steam per I.H.P. per Hour required by a Perfect Engine. Back Pressure, 1.3 Pounds.	Thermal Units per I.H.P. per Hour required by a Perfect Engine.	Decrease in Thermal Units per I.H.P. per Hour required by a Perfect Engine for each 25 Pounds Increase in Steam Pressure Expressed as a Per Cent.
I.	II.	III.	IV.	V.	VI.	VII.	VIII.
25	266.6	....	1163.28	....	39.40	36,888	....
50	297.5	30.9	1172.61	9.33	26.25	23,791	35.4
75	319.8	22.3	1179.51	6.90	21.57	19,227	19.1
100	337.6	17.8	1184.94	5.43	19.53	16,955	11.8
125	352.7	15.1	1189.47	4.53	17.54	15,185	10.4
150	365.7	13.0	1193.54	4.07	16.45	14,084	7.30
175	377.3	11.6	1197.04	3.50	15.62	13,242	5.97
200	387.8	10.5	1200.17	3.13	15.00	12,599	4.85
225	397.3	9.5	1203.14	2.97	14.51	12,067	4.22
250	406.1	8.8	1205.77	2.63	14.17	11,636	3.57
275	414.2	8.1	1208.27	2.50	13.74	11,255	3.27
3C0	421.8	7.6	1210.57	2.30	13.39	10,927	2.93

sure becomes progressively less as the pressure rises. Between 175 and 225 pounds pressure the decrease in heat consumption becomes approximately 9 per cent. The relation between steam consumption of the perfect engine and pressure is well shown by Fig. 214, which has been plotted from values given in the tables. It should be evident that, as practice moves up in the scale of pressure, the chances to save by resorting to still higher pressures become less and less.

180. The Arguments for and against the Use of Higher Pressures may be summarized as follows:

IN FAVOR OF HIGHER PRESSURES.	AGAINST HIGHER PRESSURES.
1. Smaller cylinders and consequently lighter reciprocating parts.	1. Increased weight of boiler due to thicker plates.
2. Reduced width of engine outside of cylinders.	2. Increased first cost of boiler.
3. Reduced first cost of engine.	3. Increased transportation charge due to increased weight of boiler.
4. Reduced transportation charge because of reduced weight of engines.	4. Probable increase in small heat losses from radiation and from leakage past valves and glands.
5. A possible gain in the efficiency of the engine, whereby a given power is developed on less steam and on less fuel than could have been done with a lower pressure.	5. Increased difficulty in lubrication and maintenance of packing because of higher temperature of the steam.

It is assumed that, other things being equal, the evaporative efficiency of the boiler will not be affected by such modifications in its design as are necessary to enable it to withstand the increased pressure, and hence the efficiency of the boiler does not appear on either side of the argument. The increased weight of the boiler and the decreased

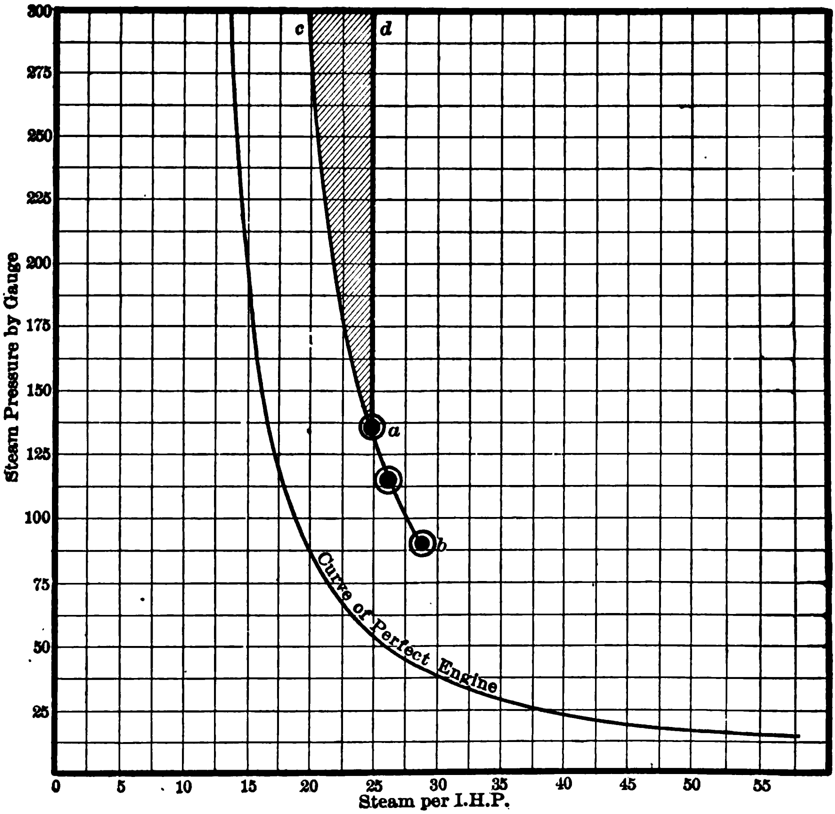


Fig. 214.

weight of the engines are not very considerable factors in the matter, and, moreover, they tend to offset one another. The leakage losses and lubrication difficulties also are not insuperable, so that the chief factor in this discussion becomes the expected increase in the economy of the engine.

181. Tests at Different Pressures.—It has been shown in the preceding discussion that the economy of a perfect or ideal engine increases with increase of pressure according to the curve plotted in

Fig. 214. An actual engine, however, may or may not show a similar increase, although it has generally been assumed that it will. The results of three tests at different pressures made upon Schenectady No. 1 are given in Table LXXVII.

Using the values of this table, points have been plotted through which the curve *ab* (Fig. 214) has been drawn. The line is a short one, and it is rather difficult to determine what the real tendency may be. It is not far from bearing a constant relation to the curve of the perfect engine. Assuming that this indication is true, and that the relation continues beyond the range of the experiments, the experimental line may be continued to the point *c*.

The relationship is important, since by it, if the steam consumption for an actual engine is known for any one pressure, its probable performance for all other pressures may be definitely calculated. The location of curve *ab* shows about 50 per cent greater steam consumption for the actual engine than for the perfect engine within the limits of the experiments. As applied to the case in hand it shows that a pressure of 150 pounds should result in a consumption of steam per I.H.P. per hour slightly in excess of 24 pounds; and that a pressure of 300 pounds should bring the consumption below 20 pounds.

182. Pressure vs. Capacity.—Thus far, consideration has been given the problem of increasing the power of a locomotive by means of higher pressures. Attention may now be given the problem of securing the same results by means of increased boiler capacity.

Table LXVIII. gives some facts concerning weights of boilers for various pressures. The figures in Col. II. were supplied by the courtesy of the Baldwin Locomotive Works. Col. III. gives the weight of the boilers of the Purdue locomotives, Schenectady Nos. 1 and 2, which are in every way similar, except as to the steam pressure, for which they were designed. From the values it will be seen that an increase in steam pressure from 150 to 240 pounds in a 60-inch boiler necessitates an increase in weight of 5900 pounds, or 18 per cent, and that an increase from 140 to 250 pounds in a 52-inch boiler necessitates an increase of 4700 pounds, or 22 per cent. It will, therefore, be sufficiently accurate for the present purpose to assume that a change of pressure from 150 to 240 pounds, an increase of 90 per cent, will demand an increase of about 20 per cent in weight of boiler.

The evaporative efficiency of a boiler has been shown to depend on the rate of evaporation at which it is worked. For Schenectady No. 1 the relation of efficiency to rate of evaporation is shown by

TABLE LXXVII.

SHOWING THE PERFORMANCE OF LOCOMOTIVE SCHENECTADY NO. 1 UNDER DIFFERENT STEAM PRESSURES.

Consecutive Number.	Laboratory Designation.	Revolutions per Minute.	Miles per Hour.	Throttle Position.	Reverse-lever, Notch Forward of Center.	Cut-off, Per Cent of Stroke.	Boiler Pressure by Gauge.	Initial Pressure from Cards.	Increase of Initial Pressure.	Percentage Increase of Initial Pressure.	Indicated Horse-power.	Increase of Power.	Percentage Increase of Power.	Pounds of Steam per Horse-power per Hour.	Decrease in Steam Consumption.	Percentage Decrease in Steam Consumption.
I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.	XIII.	XIV.	XV.	XVI.	XVII.
1	35-2-B	188.4	35.0	Wide open	2d	32	98.4	91.5	....	....	300	....	....	28.8		
2	35-2-A	190.0	35.3	"	2d	32	131.7	115.6	24.1	26	435	135	45	26.3	.5	9
3	35-2-C	189.5	35.2	"	2d	32	143.3	137.3	21.7	19	522	87	20	24.9	1.4	5

Note.—The boiler pressure for the second test is in doubt. The *initial* pressures, however, are from cards which afford ample opportunities for verification, and for this reason the initial pressure is used as a basis for comparison in the table, rather than the boiler pressure.

TABLE LXXVIII.

SHOWING CHANGE IN WEIGHT OF BOILER WITH STEAM PRESSURE.

Steam Pressure.	Weight of 60-inch Boiler.	Weight of 52-inch Boiler.
I.	II.	III.
140	......	21,035
150	33,121	......
180	35,253	......
210	38,513	......
240	39,035	......
250	......	25,775

the line *de* of Fig. 215. It will be seen that a decrease in the rate of evaporation increases the amount of water evaporated per pound of

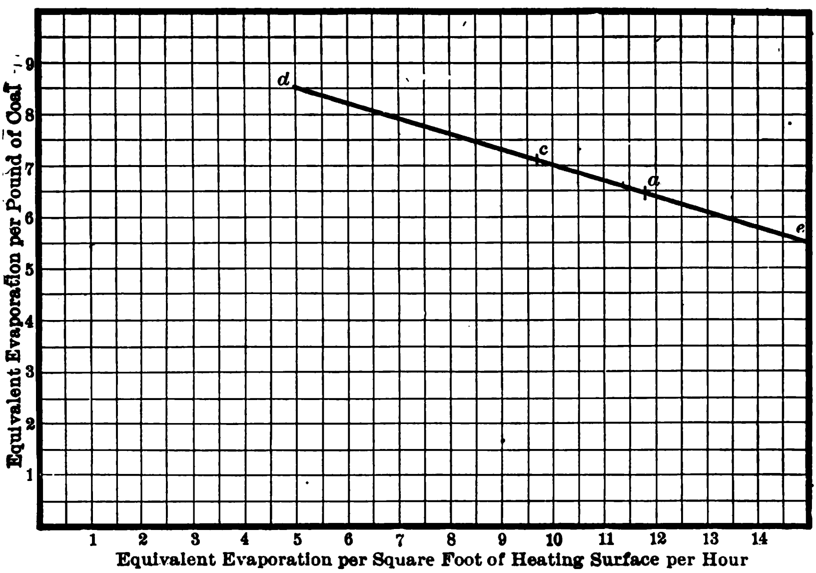


FIG. 215.—Evaporative Performance.

coal. Thus, suppose that at a certain speed and load the boiler, as at present designed, is required to evaporate 11.8 pounds of water per square foot of heating-surface per hour, its evaporative efficiency will be that represented by the point *a*. For the present purpose it may be assumed that an increase of 20 per cent in the weight of the boiler will give a proportional increase in the extent of its heating-surface, the pressure for which it is designed remaining unchanged.

On this assumption the boiler with 20 per cent increase in weight would at the same power be compelled to evaporate only 9.7 pounds of water per square foot of heating-surface, and its efficiency would rise to that represented by the point *c*, an increase in efficiency of about 11 per cent. This increase in efficiency is due wholly to the increased capacity of the boiler. Numerical values, based on the assumption that increase in capacity will be proportional to increase of weight, are given in Table LXXIX., from which it will be seen that any increase in the size of the boiler can be depended upon to yield a definite return in the improved performance of the locomotive.

TABLE LXXIX.

SAVING IN FUEL BY USING A BOILER THE CAPACITY OF WHICH IS GREATER THAN AN ASSUMED NORMAL BOILER.

Pounds of Water required to be Evaporated per Square Foot of Heating-surface per Hour in a Normal Boiler.	Percentage Saving in Fuel by Using a Boiler the Capacity of which is Greater than the Normal Boiler by 5, 10, 15, and 20 Per Cent Respectively.			
	5 Per Cent.	10 Per Cent.	15 Per Cent.	20 Per Cent.
I.	II.	III.	IV.	V.
5	0.8	1.5	2.2	2.9
6	1.0	1.9	2.8	3.7
7	1.2	2.3	3.4	4.5
8	1.5	2.9	4.1	5.3
9	1.8	3.4	4.8	6.1
10	2.0	3.9	5.5	7.1
11	2.3	4.5	6.4	8.2
12	2.7	5.1	7.4	9.4
13	3.0	5.7	8.4	10.7
14	3.4	6.4	9.4	12.0
15	3.8	7.4	10.6	13.6

183. **Summary.**—The preceding paragraphs all refer to a simple cylindered locomotive using saturated steam. They give evidence to the fact that locomotive efficiency can be increased either by employing a higher pressure or a larger boiler, and some data are presented tending to show the rate of change in each case. The problem in design which is here touched upon may be stated as follows: When there is opportunity to increase the weight of a locomotive boiler of a given class, will it be better to make a *stronger* boiler that higher steam pressure may be carried, or a *larger* boiler that the rate of evaporation may be reduced? In the one case the advantage will appear in the improved performance of the cylinders, and in the

other in the higher evaporative efficiency of the boiler. The variables in the problem have each been considered, and an attempt has been made to estimate their value, but its complete solution, which will fix limits and give a numerical measure of the benefits to be derived as the result of definite changes along either line, must await the accumulation of a greater array of facts than are at present available.*

* Since the preparation of this manuscript, there has been completed at the Purdue Laboratory (Aug. 1906), under the patronage of the Carnegie Institution, an elaborate series of tests to determine the value of high steam-pressures in locomotive service. Values which fairly represent the consumption of steam in pounds per horse-power hour under different pressures are as follows:

Boiler-pressure.	Steam per I H. P. per Hour.
120	29.1
140	27.7
160	26.6
180	26.0
200	25.5
220	25.1
240	24.7

Tests were run under the several pressures given at different speeds and cut-offs. The values quoted do not represent the minimum steam consumption nor the maximum, but are those which have been determined from a full analysis of all data. They may be accepted as fairly representative of the performance of the Purdue locomotive under normal conditions of running. A summarized statement covering the work and the results derived therefrom is as follows:

1. The results apply only to practice involving single-expansion locomotives using saturated steam.

2. Tests have been made to determine the performance of a typical locomotive when operating under a variety of conditions with reference to speed, power, and steam-pressure.

3. The rate of change in efficiency resulting from changes in steam-pressure has been established by the results of carefully conducted tests. They show that the higher the pressure the smaller the possible gain resulting from a given increment of pressure. An increase of pressure from 160 to 200 pounds results in a saving of 1.1 pounds of steam per horse-power hour, while a similar change from 200 to 240 pounds improves the performance only to the extent of .8 of a pound per horse-power hour.

4. The improvement in performance with increase of pressure will in service depend upon the degree of perfection attending the maintenance of the locomotive. The values quoted in the preceding paragraph assume a high order of maintenance. If this is lacking, it may easily happen that the saving which is anticipated through the adoption of higher pressure will entirely disappear.

5. The difficulties to be met in the maintenance both of boiler and cylinders increase with increase of pressure.

6. A simple locomotive using saturated steam will render good and efficient service when designed to work under a pressure as low as 160 pounds; under most favorable conditions no real advantage is gained by designing for pressures greater than 200 pounds.

7. Wherever the water which must be used in boilers contains foaming or scale-making admixtures, the pressure for best results should not exceed 180 pounds; where feed-water is exceptionally bad, it will be found advantageous to fix the maximum below this limit.

8. As the scale of pressure is ascended, an opportunity to further increase the weight of a locomotive should in many cases find expression in the design of a boiler of increased capacity rather than one for higher pressures.

9. For the development of a given power, any increase in boiler capacity brings its return in improved performance without adding to the cost of maintenance, or opening any new avenues for incidental losses. As a means to improvement, it is more certain than that which is offered by increase of pressure.

10. From the simple standpoint of efficiency, and neglecting all questions of maintenance above 180 pounds, it is better to utilize any allowable increase in weight by providing a larger boiler rather than to provide a stronger boiler to permit higher pressures.

11. Pressures designated in the preceding paragraphs are to be accepted as running pressures. They are not necessarily those at which safety-valves open.

12. The preceding statements justify the conclusion that steam-pressures *in* American locomotive service have already been carried to a limit which, for best results, should be accepted as maximum.

CHAPTER XXII.

CONCERNING DIAMETER OF DRIVING-WHEELS.

184. Practice with Reference to Wheel Diameters.—Prior to the renascence of the American locomotive, which had its beginning in 1894–95, the practice of this country was committed to the use of driving-wheels of comparatively small diameter. With wheels thus proportioned, locomotives were very effective at slow and moderate speeds, but at high speeds the rotation reached limits which were not equaled in the practice of other countries. Under this practice the driving-wheels of fast passenger locomotives were driven in regular and ordinary service to a speed of between 300 and 400 revolutions a minute; while a rule much employed by a prominent builder was to make the diameter of the drivers in inches equal to the speed in miles per hour for which the locomotive was designed, a rule which contemplates a speed of rotation of 336 revolutions per minute.

With the upbuilding of the modern American locomotive, driving-wheels for all classes of service have been materially enlarged, so that in ordinary service the speed of rotation is now not so high as formerly, notwithstanding which fact an analysis of the general question will not be without interest.

185. A Study Based upon Observed Facts.—From the data and discussion of Chapter V., it is evident that whenever the speed of rotation exceeds the critical speed a loss of efficiency results. If, therefore, high train speeds are demanded, the diameters of the driving-wheels should, for best results, be so proportioned as to give the desired rate of travel without exceeding the critical speed of rotation. The following illustration will serve to show how this principle may be worked out in the case of Schenectady No. 1.

The proportions of the locomotive are such that, neglecting friction, every pound of mean effective pressure exerted in the cylinders will produce a draw-bar pull of 109.4 pounds. It has already been

shown (Chapter V., paragraph 36) that the power of this locomotive becomes maximum when the speed reaches 188 revolutions per minute. Examining the conditions, first, with reference to a cut-off of 35 per cent, it will be found that the mean effective pressure at this speed is 42.4 pounds (Test 35–2–A), which is equivalent to a pull at the draw-bar of

$$42.4 \times 109.4 = 4639 \text{ pounds.}$$

If, now, it is required to operate this locomotive at a speed of 55 miles per hour, the revolutions must increase from 188 to 296 per minute, which will cause the mean effective pressure to drop to 27.4 pounds (Test 55–2–A). This makes the draw-bar pull at 55 miles

$$27.4 \times 109.4 = 2997 \text{ pounds.}$$

Suppose, now, that instead of increasing the speed of rotation from 188 to 296, the diameter of the driving-wheels were increased from 63 inches, the present diameter, to 99 inches, a diameter which will permit a speed of 55 miles an hour at 188 revolutions per minute. With these new proportions the locomotive would give but 69.4 pounds pull at the draw-bar for each pound mean effective pressure. But a speed of 55 miles would now involve only 188 revolutions per minute, and the mean effective pressure would be 42.4 pounds (Test 35–2–A), which would give a pull at the draw-bar of

$$42.4 \times 69.4 = 2943 \text{ pounds,}$$

a result which is practically identical with that obtained with the smaller wheels and a higher speed of rotation. In this case, therefore, there has been no loss or gain, so far as power is concerned when operating at a train speed of 55 miles an hour, by the substitution of 99-inch drivers for the 63-inch, which are normal to the engine.

Since, however, the engine is more economical in its use of steam at the critical speed than when the revolutions are greater, there would be some gain in mechanical action through the use of the larger driving-wheels, since, at 296 revolutions, the cylinders require 32 pounds per horse-power hour, while at 188 revolutions they require but 26.3 pounds, a gain of 21 per cent.

A similar comparison based upon a cut-off of 25 per cent gives results which are as follows:

The mean effective pressure under this cut-off is, at 188 revolutions, 29.6 pounds, and at 296 revolutions, 18.3 pounds. When the train speed is 55 miles, the draw-bar stress with the present 63-inch driving-wheels will be

$$18.3 \times 109.4 = 2002 \text{ pounds.}$$

Whereas, if the driving-wheels were increased to 99 inches the draw-bar stress would be

$$29.6 \times 69.4 = 2054 \text{ pounds,}$$

which is a gain in draw-bar stress in favor of the larger wheels. In this case, also, there would be an increase of economy resulting from the use of the larger wheels, the steam consumption under the conditions imposed by the small wheels being 30.6 pounds, and under those imposed by the larger but 27, a gain of 13 per cent.

This analysis would seem to establish the fact, within limits that are pretty well defined, the draw-bar pull at speed is not reduced by increasing the diameter of the wheels, while the cylinder action is made more efficient. Since, in the case of a locomotive, anything which saves steam may at the limit be utilized in producing more power, the benefits in increased power to be derived from an increase in the diameter of driving-wheels is drawn from a twofold source.

186. A Recapitulation of the facts of the analysis which has been given is presented in Table LXXX.

It is admitted that there are mechanical difficulties to be overcome before wheels of very large diameter can be used, also that conditions of service requ re some sacrifice of efficiency at speed to insure satisfactory performance in starting, and doubtless in many classes of service such considerations fully justify the use of the smaller wheels. The purpose is not to condemn practice, but to show what proportions are desirable for operation at speed.

Finally, in this connection it is to be noted that the modern locomotive has been given wheels which, while yet too small for highest efficiency at the rates of speed at which many locomotives are driven, are much larger than those which were used in the early '90's, when Schenectady No. 1 was built. These large-wheeled engines have proved economical in the use of water and coal.

TABLE LXXX.*

SHOWING CERTAIN RESULTS OBSERVED IN CONNECTION WITH LOCOMOTIVE SCHENECTADY, AND SIMILAR RESULTS DEDUCED FROM DATA GIVEN ON THE SUPPOSITION THAT THE DIAMETER OF ITS DRIVING-WHEELS HAD BEEN INCREASED IN THE RATIO OF 35 TO 55. SPEED CONSTANT AT 55 MILES AN HOUR. THROTTLE FULLY OPEN.

	Present Drivers, 63-inch Diameter.	Proposed Drivers, 99-inch Diameter.
Revolutions per minute	296	188
Approximate speed in miles per hour	55	55
Indicated horse-power (Table I):		
6-inch cut-off	292	298
8-inch cut-off	438	431
Tractive force, pounds:		
6-inch cut-off	1995	2054
8-inch cut-off	2987	2943
Steam per indicated horse-power per hour (Table III):		
6-inch cut-off	30.6	26.9
8-inch cut-off	32.0	26.28
Coal per indicated horse-power per hour (Table IV):		
6-inch cut-off	5.12	4.18
8-inch cut-off	6.03	4.54
Gain or loss in indicated horse-power resulting from use of 99-inch drivers in place of 63-inch drivers for speed of 55 miles an hour:		
6-inch cut-off	Gain.... 2.9 per cent	
8-inch cut-off	Loss.... 1.4 per cent	
Decrease in steam consumption resulting from use of 99-inch drivers in place of 63-inch drivers for speed of 55 miles an hour:		
6-inch cut-off	12 per cent	
8-inch cut-off	18 per cent	
Decrease in coal consumption resulting from the use of 99-inch drivers in place of 63-inch drivers for speed of 55 miles an hour:		
6-inch cut-off	18 per cent	
8-inch cut-off	23 per cent	

* This table and the arguments based thereon were first presented to the Western Railway Club, May, 1896.

CHAPTER XXIII.

ATMOSPHERIC RESISTANCE TO THE MOTION OF RAILWAY TRAINS.

187. Atmospheric Resistance.—The resistance which must be overcome by a moving train arises from several causes, as, for example, from the rolling friction of wheel on rail, the effect of gradients and curvatures in the track, the necessity of producing accelerations in the speed, the friction of journals, and the resistance of the atmosphere.

The work which must be done to overcome the effect of grades and to produce accelerations in speed can be accurately determined, and the value of journal and rolling friction, when considered apart from complicating conditions, is already somewhat definitely known, but the available evidence concerning atmospheric resistance is contradictory and the result of its application uncertain. This fact makes of interest certain experiments, which were conducted at the Engineering Laboratory of Purdue University during the school year 1895–96 by Professor H. C. Solberg, then a graduate student in the laboratory.*

The conditions under which experiments were made were assumed to be similar to those surrounding a train moving through still air, and the object of the experiments has been to disclose the value of forces resulting from the resistance offered by a quiescent atmosphere

* An account of these experiments was first published as a paper before the Western Railway Club, April, 1896. The principal references given were:

"A Study of Atmospheric Resistance to the Motion of Railway Trains." A thesis by H. C. Solberg, B.S., M.E., '96.

"A Study of Air-currents in a Rectangular Conduit." A thesis by Augustus C. Spiker, B.S.,'96.

"An Investigation of the Air-currents about a Moving Car or Train of Cars." A thesis by Norman E. Gee, B.S., '96.

to the forward movement of trains through it. No attempt was made to consider the effects resulting from oblique or other winds.

188. The Plan of the Experiments involved a rectangular conduit, within which a current of air having any desired velocity could be maintained. Within this conduit, and exposed to the action of the air-currents, small dummy or model cars were mounted. Each model was connected by means of a sensitive dynamometer, with a suitable base so arranged as to indicate the value of any force tending to displace it in the direction of its length. A single model, or any number of models placed in order, as in a train, could be employed in any given experiment, the effect of the wind upon each car being always shown by the indication of its attached dynamometer. It is evident that, as a matter of principle, it is not material whether the model is at rest and the air is moved past it, or the air still and the model moved through it; that is, if the velocity of movement is the same in each case the value of the reaction between the wind and model will be the same. While the effect of the actual conditions surrounding the apparatus employed will be carefully reviewed in another paragraph (see paragraph 203), it may for the present be assumed that effects observed under the conditions of the experiments are the same as would have been observed had the model cars been caused to move through still air. Before proceeding the details of the apparatus employed should be briefly examined.

189. Conduit.—The conduit in which the flow of air was maintained for the experiments is in the form of a rectangular tube 20×20 inches in section and 60 feet in length. A cross-section is shown by Fig. 216. The lower face is of solid wood; the upper, also of wood, is pierced at intervals of six feet by good-sized openings, through which one may reach into the interior. These openings are closed by tight-fitting covers. The side faces of the conduit consist of large panels of glass set in wooden frames. The glass sides expose to view the whole interior of the conduit, so that both the position of the model cars and the reading of their dynamometers can readily be seen by the observer on the outside. The conduit is practically air-tight, the joints between glass and wood being covered with glued strips of paper. The interior surfaces also are unbroken from end to end, and, where of wood, are made so smooth by shellac as to offer but slight resistance to the passage of air through the tube.

190. Air-supply.—The conduit is connected at one end with a No. 60 Sturtevant blower, the opposite end being open to the labora-

tory. The whole apparatus being in one room, the duty of the blower is simply that of circulating the air of the room through the tube, forcing it in at one end, and allowing it to discharge at the other. The blower is of sufficient power to produce air-currents in the conduit having a velocity of 100 miles an hour.

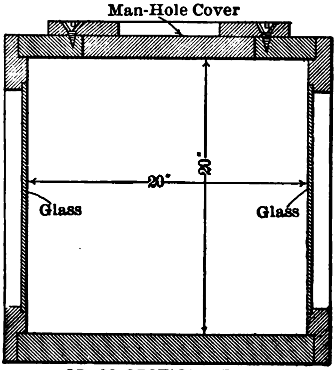


FIG. 216.

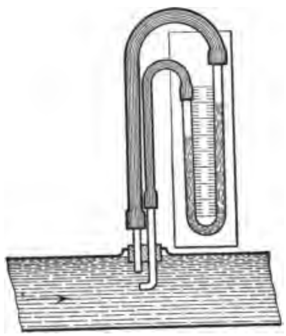


FIG. 217.

191. The Determination of the Velocity of the Air-currents.— The velocity of the moving air within the conduit was determined by use of instruments in the form of Pitot's tubes.* These were

* A simple form of Pitot tube is shown by Fig. 217. It consists of two small tubes, having ends inserted into the flowing stream, the velocity of which it is desired to measure. The end of one tube is shaped to face the flow of the stream, while that of the other is normal to the flow. The exposed ends of the tubes connect with a U-shaped glass tube, partially filled with water or other liquid. It will be seen that as both sides of the U tube are in connection with the flowing stream, the difference in the height of liquid columns in the U tube cannot be due to the pressure of the flowing stream, but must result from the motion of the stream.

The relation between the displacement of the gauge and the velocity of the flowing stream is expressed by the equation

$$v^2 = 2gh,$$

when v is the velocity of the stream in feet per second, and h the difference in height of the gauge columns in feet, measured in terms of a substance having the same density with the one whose velocity is to be determined.

Many experimenters have from time to time testified as to the accuracy of this method of measuring velocities. It has long been used by physicists and meteorologists, and Professor W. S. Robinson, who has recently employed it extensively in determining the flow of natural gas in pipes, states that he was able to check his results with those obtained from meters with a satisfactory degree of certainty.

In the present experiments water was used in the U tubes, and the relation

made up of two brass tubes, arranged within a larger tube or jacket, all being cemented together by resin, which filled the interior of the jacket around the smaller tubes. The interior diameter of the small tubes was a sixteenth of an inch, and the diameter of the jacket-tube somewhat less than a half-inch, while the length of the combination was such as made it possible to reach from the exterior to any portion of the interior of the conduit. This portion of the apparatus is shown by Fig. 218. When in use the tip end, *a*, of the gauge was inserted into the current through holes bored in the top planking, a cork bushing lining the hole, and making tight the joint between the wood and the gauge. Each of the two small brass tubes making up a gauge was then connected by rubber tubing with one side of a glass U tube fixed to a suitable scale outside of the conduit. The U tubes were sealed with water, from the displacement of which the velocity of the air passing the tips of the gauge was determined. Five such gauges with all their connections are shown in place in the conduit by Fig. 219.

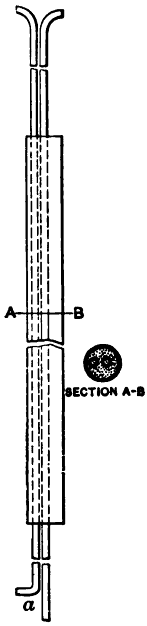


FIG. 218.

The several gauges employed were subjected to a careful examination, involving a series of simultaneous observations in connection with a systematic interchange of position, to determine whether all could be depended upon to give like indications when the conditions were the same.

Another preliminary to the main investigation was that of determining the relative velocity of the stream of air at different points in the cross-section of the conduit. This was done by dividing the

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between the density of water and air is such as to make a column of water one inch high the equivalent of a column of air 68.37 feet high. The equation therefore becomes

$$v^2 = (2 \times 32.2 \times 68.37)h = 4403h,$$

where v is the velocity in feet per second and h is the head in inches of water. If, therefore, the U tube of a gauge used in the experiment showed a displacement of an inch and a half, the velocity of the air passing the tips of gauge in feet per second was assumed to be

$$v = \sqrt{4403 \times 1.5} = 81.2.$$

cross-section into twenty-five or more imaginary squares, and by observing the velocities at the center of several of them at the same instant, after which some of the gauges were changed to other squares and the process repeated, the observations for each set of readings overlapping those of the preceding set as a check on the constancy of conditions. A number of typical diagrams resulting from this process are presented as Fig. 220. They show velocity of the cur-

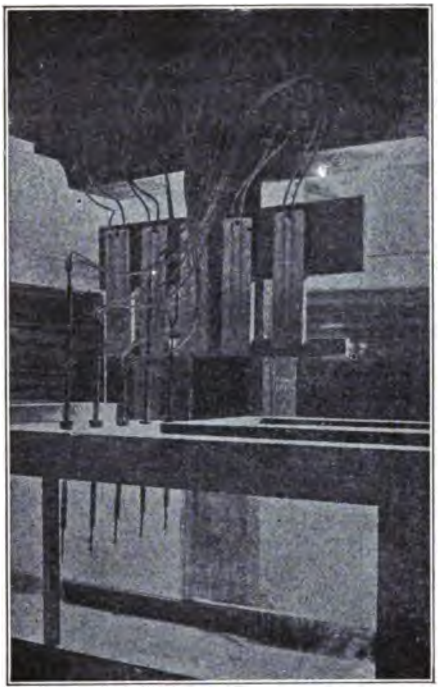

FIG. 219.

rent in miles per hour for different portions of the cross-section of the conduit.

That there might be no uncertainty, also, as to the character of the flowing current of air, the cross-section of the stream was carefully examined at many points throughout the length of the conduit, and as a result the following conclusions were reached:

1. That while considerable unevenness of flow was observed near the initial end of the conduit the eddies disappeared at a distance of 35 feet from the initial end, and from this point to a point near the

discharge end of the conduit the flow was found to follow lines which were approximately straight.

2. That the glass surfaces forming the sides of the conduit offered less resistance to the movement of the air than the wooden surfaces forming the top and bottom.

3. That the lowest velocities were found, as would be expected, in the corners of the conduit, that is, where the sides joined with the top and bottom.

Wood

	21	23	24	23	21	
	25	29	29	29	25	
Glass	25	29	29	29	25	Glass
	25	28	29	29	25	
	21	23	25	23	21	

Wood

Wood

	47	53	53	53	46	
	54	61	61	61	54	
Glass	55	61	61	61	55	Glass
	55	61	61	61	54	
	46	52	53	52	47	

Wood

Wood

	80	95	95	96	81	
	96	100	100	100	96	
Glass	96	100	100	100	96	Glass
	96	100	100	100	96	
	81	95	95	95	81	

Wood

FIG. 220.—Relative Velocities in Different Portions of the Conduit.

4. That there was a comparatively large vein in the interior of the stream, all portions of which flowed with practically the same velocity.

The experiments which are to be described made use of that portion of the stream which was most free from eddies, and which was least influenced by the walls of the conduit.

192. The Model Cars.—Having obtained means for making a

breeze of satisfactory quality and for determining its velocity, the next and last step concerned the model cars which were to be exposed to its influence. To facilitate the description these model cars will hereafter be referred to as *models.* These were $\frac{1}{32}$ the size of an assumed standard box car, the body of the model extending downward and occupying the space which in an actual car intervenes

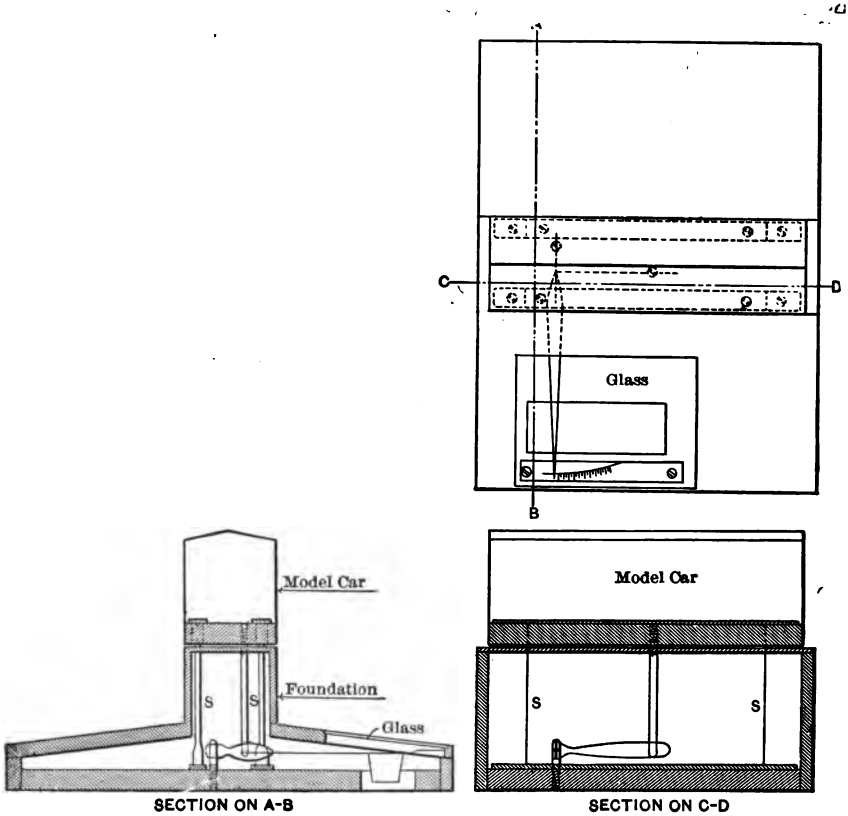

FIG. 221.

between the sills and the rails. Each model was $12\frac{1}{16}$ inches long, $3\frac{3}{8}$ inches wide, and $4\frac{1}{2}$ inches high. Its form may be more perfectly apprehended by reference to the drawing (Fig. 221). The painted tin body of the model was fitted over a wooden base supported by four leg-pieces of light, hard-rolled sheet brass *S*, which in turn were securely fastened to a suitable foundation.

The length and lightness of these legs, or springs, allowed the car to be displaced longitudinally, under the action of the slightest force, and they were at the same time so proportioned as to resist all tendency to motion in other directions. Between the body of the car and its foundation, also, and entirely independent of the springs already referred to, was a system of levers, the purpose of which was to multiply any longitudinal displacement to which the model might be subject. These levers were made of thin metal, the several parts being soldered to each other. All motion, consequently, was within the elastic limit of the parts affected. There were no loose joints. The whole arrangement proved to be both sensitive and reliable. The least pressure upon the car would result in a movement of the pointer, and the pointer would promptly return to its zero when the force producing the displacement had ceased to act. Excessive vibrations of the pointer were prevented by a vertical fin which could be made to dip into light oil contained in a suitable pan beneath. That no part of the dynamometer might be directly affected by the currents of air acting upon the model, the mechanism was entirely enclosed in the foundation, a portion of the surface of which was of glass through which the movement of the pointer could be observed.

The degree of refinement attending the action of these dynamometer cars will be appreciated when it is said that, while the actual movement of the car was always slight, the leverage was such that an inch and a quarter movement of the pointer was readily obtained. The springs for a number of cars were made so flexible as to give an inch movement of the pointer under the force of one ounce acting upon the end of the model. Two models, however, to serve at the ends of trains, were provided with much stiffer springs.

The foundations were longer than the models, so that when arranged in trains the proper spacing of models would be observed, while the foundation would present an unbroken surface. Sections of blank foundation, also, were supplied in front and rear of train. A drawing showing a train of three cars arranged in the conduit is shown by Fig. 222, and a front view of a train taken from within the conduit is given as Fig. 223.

193. **Observations.**—With the desired number of models arranged as a train, and with a single Pitot tube located at A (Fig. 222), which alone was used in determining velocities in the conduit, the experiments proceeded about as follows: The blower engine was started and

allowed to run at a slow speed for a sufficient time to secure constancy of conditions with the conduit, after which readings were taken simultaneously from the gauge *A* and the dynamometers of the several cars composing the train. These observations were repeated at intervals of thirty seconds until five readings had been taken, when the averages of the five sucessive readings were brought forward to a condensed log of observation. When one set of readings had been taken, the speed of the blower was increased, and all observations made for the new conditions. In this manner the work was advanced with each length of train, the velocities of the air-currents varying from 20 miles per hour to something over 100 miles per hour. No effort was made to obtain definite conditions of air-velocity, the

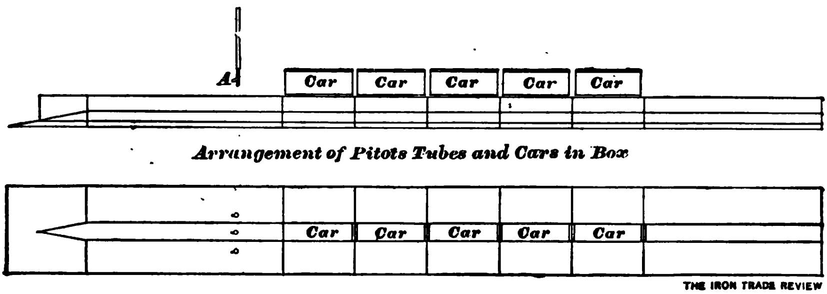

FIG. 222.—Side View of Train in Conduit.

object being to have a constant flow, and to observe accurately what were the precise values by which the conditions were defined.

The work extended through several weeks, and the care taken throughout its progress was such that the data are remarkably consistent. Tables LXXXI. to LXXXVI., which are given herewith, are somewhat changed in form, and are much condensed from the very elaborate presentation by Professor Solberg.

194. **Observed and Calculated Results.**—The significance of the seven different headings appearing in the several tables accompanying may be explained as follows:

I. *Gauge Displacement, Inches of Water.*—Values in this column are results obtained from direct readings of the U tube gauges connected with the Pitot tubes. They represent pressures measured in inches of water, due to the velocity of the air-currents within the conduit.

II. *Pressure Equivalent of Gauge Displacement in Pounds per Square Foot.*—These values are calculated directly from those of the preceding column, on the assumption that a cubic foot of water weighs 62.3 pounds. Thus, a column of water an inch high is the equivalent of

$$\frac{1\times 62.3}{12}=5.19 \text{ pounds per square foot,}$$

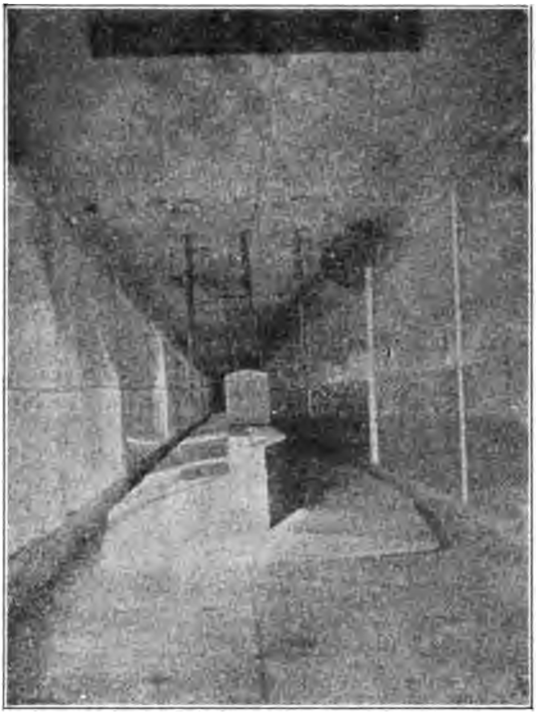

FIG. 223.

so that the values of Col. II. are equal to those given in Col. I. multiplied by 5.19. The values given in this column will be found useful for purposes of comparison, since wind pressures are usually measured in pounds per square foot.

III. *Pressure by Gauge, Multiplied by the Area of the Cross-section of the Model in Square Feet.*—In general, this would be expected to give the force with which the wind would act upon the end of a model. The cross-section of each model was equal to 14 square inches, or

TABLE LXXXI.

ONE MODEL.

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Pounds per Square Foot.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Feet.	Actual Force Tending to Displace Model, as Shown by Model Dynamometer in Pounds.	Ratio of Force Tending to Displace Model (Col. IV) to Pressure Due to Velocity (Col. III).	Velocity Calculated from Reading of Gauge. Feet per Second.	Velocity Calculated from Reading of Gauge. Miles per Hour.
	I.	II.	III.	IV.	V.	VI.	VII.
1	0.3	1.6	.15	.094	.61	36	25
2	0.8	4.2	.40	.219	.54	60	41
3	2.6	13.5	1.31	.625	.48	107	73
4	3.8	19.7	1.91	.906	.47	130	88
5	5.0	26.0	2.52	1.250	.49	149	102

TABLE LXXXII.

TWO MODELS.

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Pounds per Square Foot.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Ft.	Actual Force Tending to Displace Model, as Shown by Model Dynamometer in Pounds.			Ratio of Force Tending to Displace Models (Col. IV) to Pressure Due to Velocity (Col. III).			Velocity Calculated from Reading of Gauge	
				IV.			V.			Feet per Second	Miles per Hour.
	I.	II.	III.	First Model.	Second Model.	Both Models.	First Model	Second Model.	Both Models	VI.	VII.
1	0.3	1.6	.15	.047	.031	.078	.31	.21	.52	36	25
2	1.0	5.2	.50	.200	.078	.278	.40	.15	.55	66	45
3	2.1	10.9	1.06	.438	.150	.588	.41	.14	.55	96	66
4	2.8	14.6	1.42	.563	.172	.735	.40	.12	.52	111	76
5	5.4	28.0	2.72	1.094	.328	1.422	.40	.12	.52	154	105

[illegible]	I	II	III	IV				V				VI	VII
				[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
1	.66	3.4	.33	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
2	1.6	8.3	.81	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
3	2.5	13.0	1.20	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
4	4.2	21.8	2.11	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5	5.6	20.1	2.82	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE LXXXIV.

FIVE MODELS.

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Pounds per Square Foot.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Ft.	Actual Force Tending to Displace Model, as Shown by Model Dynamometer in Pounds.						Ratio of Force Tending to Displace Model (Col. IV) to Pressure Due to Velocity (Col. III).						Velocity Calculated from Reading of Gauge.	
				First Model.	Second Model.	Third Model.	Fourth Model.	Fifth Model.	Five Models.	First Model.	Second Model.	Third Model.	Fourth Model.	Fifth Model.	Five Models.	Feet per Second.	Miles per Hour.
	I.	II.	III.	IV.						V.						VI.	VII.
1	1.25	6.5	.63	.234	.016	.028	.025	.094	.397	.37	.025	.044	.039	.15	.63	74	51
2	2.3	12.0	1.16	.469	.031	.047	.041	.156	.744	.40	.026	.040	.035	.13	.63	101	69
3	3.9	20.2	1.96	.750	.059	.069	.069	.219	1.166	.38	.030	.035	.035	.11	.59	131	89
4	4.8	24.9	2.42	.938	.075	.100	.094	.275	1.482	.39	.031	.041	.038	.11	.61	145	99
5	5.4	28.0	2.72	1.063	.088	.109	.106	.297	1.663	.39	.032	.039	.039	.11	.61	154	105

TABLE LXXXIII.

THREE MODELS

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Pounds per Square Foot.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Ft.	Actual Forces Tending to Displace Model, as Shown by Model Dynamometer in Pounds.				Ratio of Force Tending to Displace Model (Col. IV) to Pressure Due to Velocity (Col. III).				Velocity Calculated from Rea·ing of Gauge	
												Feet per Second.	Miles per Hour.
	I.	II.	III.	IV.				V.				VI.	VII.
				First Model.	Second Model.	Third Model.	Three Models.	First Model.	Second Model.	Third Model.	Three Models.		
1	.65	3.4	.33	.125	.031	.056	.212	.38	.09	.17	.64	54	37
2	1.6	8.3	.81	.325	.017	.109	.451	.40	.02	.13	.53	84	57
3	2.5	13.0	1.26	.500	.063	.163	.726	.40	.05	.13	.58	105	72
4	4.2	21.8	2.11	.844	.109	.250	1.203	.40	.05	.12	.57	136	93
5	5.6	29.1	2.82	1.125	.125	.328	1.578	.40	.04	.12	.56	157	107

Table LXXXIV.

FIVE MODELS.

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Pounds per Square Foot.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Ft.	Actual Force Tending to Displace Model, as Shown by Model Dynamometer in Pounds.						Ratio of Force Tending to Displace Model (Col. IV) to Pressure Due to Velocity (Col. III).						Velocity Calculated from Reading of Gauge.	
				First Model.	Second Model.	Third Model.	Fourth Model.	Fifth Model.	Five Models.	First Model.	Second Model.	Third Model.	Fourth Model.	Fifth Model.	Five Models.	Feet per Second.	Miles per Hour.
	I.	II.	III.	IV.						V.						VI.	VII.
1	1.25	6.5	.63	.234	.016	.028	.025	.094	.397	.37	.025	.044	.039	.15	.63	74	51
2	2.3	12.0	1.16	.469	.031	.047	.041	.156	.744	.40	.026	.040	.035	.13	.63	101	69
3	3.9	20.2	1.96	.750	.059	.069	.069	.219	1.166	.38	.030	.035	.035	.11	.59	131	89
4	4.8	24.9	2.42	.938	.075	.100	.094	.275	1.482	.39	.031	.041	.038	.11	.61	145	99
5	5.4	28.0	2.72	1.063	.088	.109	.106	.297	1.663	.39	.032	.039	.039	.11	.61	154	105

Table LXXXV.

TEN MODELS.

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Lbs. per Sq. Ft.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Ft.	Actual Force Tending to Displace Model, as Shown by Model Dynamometer in Pounds.											Ratio of Forces Tending to Displace Model (Col. IV) to Pressure Due to Velocity (Col. III).											Velocity Calculated from Reading of Gauge.	
	I.	II.	III.	IV.											V.											VI. Feet per Second.	VII. Miles per Hour.
				First Model.	Second Model.	Third Model.	Fourth Model.	Fifth Model.	Sixth Model.	Seventh Model.	Eighth Model.	Ninth Model.	Tenth Model.	Ten Models.	First Model.	Second Model.	Third Model.	Fourth Model.	Fifth Model.	Sixth Model.	Seventh Model.	Eighth Model.	Ninth Model.	Tenth Model.	Ten Models.		
1	.5	2.6	.25	.094	.006	.008	.009	.009	.009	.009	.009	.009	.022	.184	.42	.027	.035	.040	.040	.040	.040	.040	.040	.098	.82	47	32
2	.65	3.4	.33	.131	.009	.017	.016	.015	.015	.015	.015	.016	.034	.283	.40	.027	.051	.049	.045	.045	.045	.045	.049	.103	.86	54	37
3	.9	4.7	.45	.175	.013	.016	.017	.017	.016	.016	.018	.016	.044	.348	.38	.029	.035	.037	.037	.035	.035	.039	.035	.097	.76	63	43
4	1.1	5.7	.55	.219	.016	.022	.021	.019	.020	.021	.021	.020	.056	.435	.34	.024	.034	.032	.029	.030	.032	.032	.030	.086	.67	70	48
5	1.5	7.8	.76	.313	.021	.033	.034	.035	.035	.034	.036	.034	.088	.663	.41	.028	.043	.045	.046	.046	.045	.047	.045	.116	.87	81	55
6	2.0	10.4	1.01	.406	.029	.038	.038	.040	.038	.039	.040	.039	.103	.810	.40	.029	.038	.038	.040	.038	.039	.040	.039	.103	.80	94	64
7	2.25	11.7	1.13	.469	.031	.047	.045	.046	.046	.046	.047	.047	.115	.939	.41	.027	.041	.039	.040	.040	.040	.041	.041	.101	.82	100	68
8	3.35	17.4	1.69	.656	.048	.060	.060	.062	.062	.062	.061	.063	.163	1.297	.38	.028	.035	.035	036	.036	.036	.036	.037	.095	.75	121	83
9	3.8	19.7	1.91	.750	.061	.075	.075	.075	.075	.075	.075	.075	.181	1.517	.38	.032	.039	.039	039	.039	.039	.039	.039	.094	.78	129	88
10	4.1	21.3	2.07	.813	.069	.081	.079	.081	.080	.080	.081	.081	.200	1.645	.39	.033	.039	.038	039	.038	.038	.039	.039	.097	.79	134	91
11	4.4	22.8	2.21	.875	.072	.088	.090	.088	.089	.088	.090	.089	.213	1.782	.39	.032	.039	.041	.039	.040	.039	.041	.040	.096	.80	139	95
12	4.8	24.9	2.42	.938	.075	.094	.094	.094	.094	.094	.094	.093	.244	1.914	.39	.031	.039	.039	.039	.039	.039	.039	.038	.101	.79	145	99
13	5.2	27.0	2.62	1.031	.081	.098	.097	.097	.098	.098	.097	.098	.256	2.051	.39	.031	.038	.037	.037	.038	.038	.037	.038	.097	.75	151	103
14	5.4	28.1	2.73	1.125	.088	.109	.110	.110	.111	.111	.111	.110	.265	2.250	.41	.032	040	.040	.040	.041	.041	.041	.040	.097	.82	154	105

TABLE LXXXVI.

TWENTY-FIVE MODELS.

Number of Test.	Gauge Displacement, Inches of Water.	Pressure Equivalent of Gauge Displacement, Pounds per Square Foot.	Pressure by Gauge Multiplied by Area of Cross-section of Model in Square Ft.	Actual Force Tending to Displace Model, as Shown by Model Dynamometer in Pounds.								
	I.	II.	III.	IV.								
				First Model.	Second Model.	Third Model.	Ninth Model.	Fifteenth Model.	Nineteenth Model.	Twentieth Model.	Twenty-fifth Model.	All Twenty-five Models.
1	2.1	10.9	1.06	.422	.031	.047	.041	.044	.041	.044	.109	1.503
2	2.5	13.0	1.26	.500	.041	.056	.050	.050	.050	.050	.125	1.772
3	3.0	15.6	1.51	.594	.047	.059	.056	.053	.059	.059	.156	2.041
4	3.6	18.7	1.81	.719	.056	.069	.069	.063	.069	.075	.184	2.471
5	4.1	21.3	2.07	.812	.063	.081	.088	.081	.075	.081	.213	2.888
6	4.3	22.3	2.16	.875	....	.088	.088	.088	.088	.088	.219	.....
7	4.8	24.9	2.42	.938	....	.094	.094	.094	.094	.094	.244	.....
8	5.1	23.5	2.57	1.000	....	.100	.100	.100	.100	.100	.263	.....

Number of Test.	Ratio of Forces Tending to Displace Model (Col. IV) to Pressure Due to Velocity (Col. III).									Velocity Calculated from Reading of Gauge. Feet per Second.	Velocity Calculated from Reading of Gauge. Miles per Hr.
	V.									VI.	VII.
	First Model.	Second Model.	Third Model.	Ninth Model.	Fifteenth Model.	Nineteenth Model.	Twentieth Model.	Twenty-fifth Model.	All Twenty-five Models.		
1	.39	.029	.044	.038	.041	.038	.041	.102	1.40	96	66
2	.39	.032	.044	.039	.039	.039	.039	.099	1.38	105	72
3	.39	.031	.039	.037	.035	.039	.039	.103	1.47	115	78
4	.39	.031	.038	.038	.034	.038	.041	.102	1.35	126	86
5	.39	.030	.039	.042	.039	.036	.039	.103	1.39	134	92
6	.40	....	.041	.041	.041	.041	.041	.101	.....	138	94
7	.39	....	.038	.038	.039	.039	.039	.101	.....	145	98
8	.39	....	.039	.039	.039	.039	.039	.102	.....	150	102

.097 of a square foot, and as the pressure per square foot is given by Col. II., the values of Col. III. are found by multiplying those of Col. II. by .097.

IV. *Actual Force in Pounds Tending to Displace Model, as Shown by Model Dynamometer.*—In all excepting Table LXXXI. more than one column appears under this numeral. The values are, in every case, those which have resulted from direct readings of the model dynamometer. When compared with those of Col. III. they show the effect of the form of the model in modifying the pressure resulting from the moving current of air when a single model is involved, or the effect of its position in the train when several are employed.

V. *Ratio of Actual Forces Tending to Displace Model, or Models, to Pressure Due to Velocity.*—These values are obtained by dividing the several values given under IV. by the corresponding values given in Col. III. They represent the fraction of the pressure due to the velocity of the moving air, which appears as an actual force tending to displace the several models.

VI. *Velocity of Air-currents Calculated from Gauges in Feet per Second.*—The velocities were calculated by use of the equation

$$v^2 = 4403h,$$

where v is the velocity in feet per second, and h is the displacement of the water in the gauge measured in inches.*

VII. *Velocity of Air-current Calculated from Reading of Gauge in Miles per Hour.*—These values are deduced directly from those of the preceding column.

195. One Model.—The effect of a current of air, impinging directly upon the end of a single model, may be assumed to represent the sum of three partial effects: (1) The effect of the direct action due to the exposure of the initial end of the model; (2) the effect of fric-

* It may be noted, also, that if the velocity is expressed in miles per hour, and the head in terms of pressure in pounds per square foot, this equation may readily be reduced to

$$P = .0025V^2,$$

which, therefore, like the equation to which this note refers, expresses a general relationship existing between velocity of air and resulting pressure. The equation is one often proposed and sometimes used as a means for determining wind pressures on structures, but the form of structures so modifies the pressure effects produced by wind that the equation is really useful only for the purpose of determining velocities.

tional action along the sides and top of the model; and (3) the effect of diminished pressure, or "suction," at the rear of the model.

It is significant that the numerical value of the sum of these effects upon the model is much less than the calculated value based upon the cross-section of the model, and the indications of the pressure-gauge. Thus, by Table LXXXI. the first test shows that the gauge displacement (Col. I) was .3 of an inch, which is equivalent to a pressure per square foot of 1.6 pounds (Col. II) or to a pressure of 0.15 of a pound upon an area equal to that of the cross-section of the model (Col. III), whereas the actual force tending to displace the model, as shown by its attached dynamometer, was but .094 of a pound (Col. IV); that is, the sum total effect of the wind upon the model is but 61 per cent (Col. V) of the calculated force based upon the area of its exposed or cross section. The wind velocity for this experiment was equal to 36 feet per second (Col. VI), or 25 miles per hour (Col. VII).

The last experiment recorded in the same table shows the gauge displacement to have been 5 inches of water, which is equivalent to a pressure of 26 pounds per square foot, which pressure, acting upon an area equal to the cross-section of the model, would be expected to result in a force of 2.52 pounds, whereas the actual force tending to displace the model, as shown by its attached dynamometer, was but 1.25 pounds or 49. per cent of the calculated force, based upon the cross-section of the model. The velocity of the current in the last experiment was 102 miles an hour.

A review of all the figures presented in this table will show that, in every case, the force tending to displace the model is less than that found by multiplying the calculated wind pressure of unit area by the area of the cross-section of the model. The value of the ratio, while nearly constant, tends to become less as the velocities of the air-currents are increased. The error would not be great if the ratio of the actual force to the calculated force were assumed to be always .5.

It is an interesting fact that the direct pressure on the front of the model, the friction of the wind along its sides and top, and the suction at its rear, taken altogether, should actually be of less value than that which results from the impinging stream of air on the point of the gauge, but it is one that is well established.*

* Two important facts concerning pressures resulting from air-currents are: First, that the total pressure upon planes of different areas is not necessarily proportional to the area of the exposed surface; and, secondly, that the total pressure upon a flat surface constituting one face of a solid body is greatly affected by the form cf

196. Two Models.—When two models are arranged in a train, the first is affected by the direct force of the wind, while the second is affected by the suction of the passing stream, and both are influenced by the frictional effects of the wind upon sides and top. The results of experiments upon two models are given in Table LXXXII., in which, under Cols. IV. and V., the effects upon the separate models, and upon both models taken together, are given.

In reviewing the first experiment, as presented in this table, it will be seen that, while the calculated pressure acting upon an area equal to that of the cross-section of the train is .15 of a pound, the sum of the readings of the dynamometer for both models shows but .078 of a pound or 52 per cent of the calculated amount. This is but a trifle more than was found for a single model. An examination of the table will show also that the dynamometer readings of the first model were less than those observed when a single model was exposed to the influence of the air-currents (Table LXXXI). This result is due to the fact that the second model removed from the first the effect of the suction influences. The results show that the force acting upon the first model was about .40 of the calculated force; that acting upon the second model about .14 of the calculated force; and that acting upon the two models together about .54 of the amount calculated, which values are to be compared with the .50 shown for one model. Doubling the length of the train resulted in this case in an increase of force in the ratio approximately of .50 and .54, that is, in an increase of about 8 per cent.

197. Trains of Three, Five, Ten, and Twenty-five Models.—The results of experiments upon trains, varying in length from three models to twenty-five models, appear in Tables LXXXIII. to LXXXVI. inclusive.

198. The First Model of a Train.—In all of these cases it will be seen that the forces acting upon the first model are practically the same whenever the velocity of the current is the same. The conclusion, therefore, seems to be justifiable that whenever a train is composed of more than two models, the resistance of the first model is a function of the velocity of the air-current only. This statement is, perhaps, not absolutely true, but it is practically so. Again, the value of the force is, approximately, .4 of the calculated force, based

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the solid. Little has been done as yet to define exact relationships arising from these conditions.

upon the pressure equivalent of the velocity of the wind as disclosed by gauge, and an area equal to that of the cross-section of the model.

199. The Last Model of a Train.—The forces to be resisted by the last model become less as the length of the train is increased, a condition doubtless due to the fact that the enveloping layers of air immediately about the train, and which are affected by frictional contact with it, become thicker and thicker in passing from the front to the rear, with the result that in a long train the currents immediately about the last model are less active than when the train is shorter, and as a consequence the suction effect is reduced. The data show that with the two-model train the rear model resists a force which is 14 per cent of the calculated pressure, based upon the velocity of the current and the area of the cross-section of the train; with the three-model train it is 13 per cent; with the five-model train it is 12 per cent; with trains of ten models in length it is less than 10 per cent; but with a train of twenty-five models it is still about 10 per cent.

200. The Second Model of a Train.—In all experiments, when more than two models composed the train, the forces acting upon the second model of a train appear to have been less than those acting upon any other model of the train. This is explained on the assumption that the currents in passing the first model are so deflected that some of the wave-like lines pass around the second model, thus relieving it of a portion of the force to which it would otherwise be subjected.

201. Models between the Second and the Last of a Train.—Whatever the length of the train, all intermediate models, the second excepted, seem to have been met by an equal force regardless of their location in the train. Thus, with a ten-model train and a wind velocity of 64 miles per hour, the observed force in pounds acting upon the several models from the third to the ninth inclusive was .038, .038, .040, .038, .039, .040, and .039 respectively. For all experiments the percentage of the calculated pressure, based upon wind velocity and cross-section of the train, which appears as a force acting upon intermediate cars, is shown to be between 3.8 per cent and 4 per cent. This, of course, is the sum of frictional action along sides and top, and such effect as may arise from eddies between the models.

202. Distribution of Forces Acting throughout the Length of the Model Train.—The preceding paragraphs show that each portion of a train of models presents a resistance to the currents of air moving past it, which is a fixed percentage of the pressure equivalent of the

velocity of the current; or, to make the statement more concise, the resistance offered by each portion of the train is a constant function of the velocity of the current. This relationship is shown graphically for a train of ten models by Fig. 224. It holds good for all velocities.

203. Relation of Force and Velocity.—The relation between the velocity of the current and the resulting forces acting upon each model of a train may be shown by plotting the dynamometer readings (Col. IV) for each of the several models, with the velocities (Col. VII)

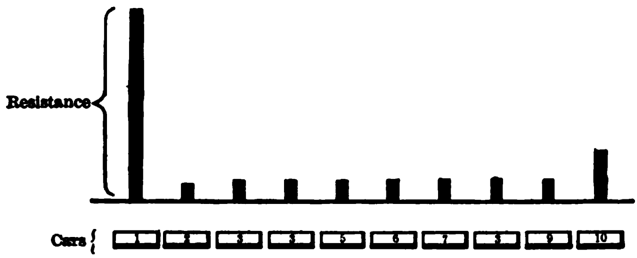

FIG. 224.—Relative Resistance Offered by the Several Cars of a Train.

corresponding. From a smooth curve drawn through the points thus obtained, equations may be written to represent the velocity, as follows:

For a single model alone

$$a_1 = .000116V^2, \quad \ldots \ldots \ldots \quad (22)$$

for the first model of a train

$$a_f = .000097V^2, \quad \ldots \ldots \ldots \quad (23)$$

for the last model of a train

$$a_l = .000025V^2, \quad \ldots \ldots \ldots \quad (24)$$

for the second model of a train

$$a_s = .000008V^2, \quad \ldots \ldots \ldots \quad (25)$$

for any intermediate model of a train

$$a_i = .000010V^2. \quad \ldots \ldots \ldots \quad (26)$$

In the preceding equations a is force in pounds acting upon the model in the direction of its length, and V is velocity of the air-current in miles per hour. In the form in which they are given the

equations are not of general application, since they are based on the dimensions of the particular models employed in the experiments.

Equations of a more general character may, however, be readily obtained by reducing the observed forces acting upon each model to equivalent forces which would have been observed had the area of the cross-section of the models been one square foot, the proportions of the models remaining unchanged. Thus, the area of the cross-section of the actual models was .097 of a foot. By dividing the

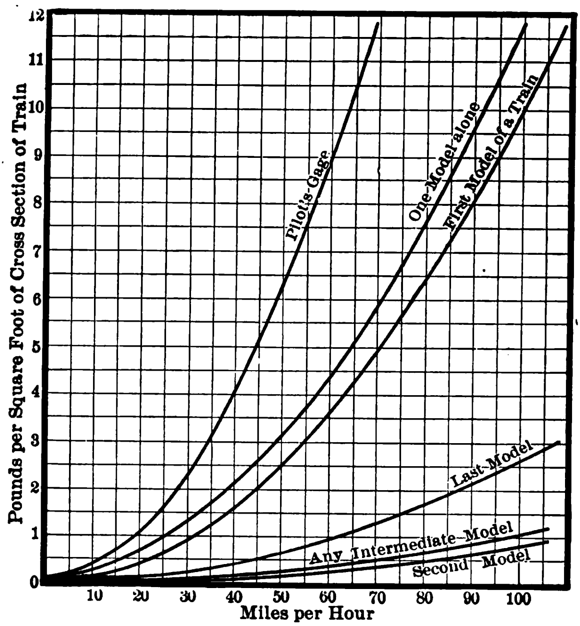

FIG. 225.—Relation of Force and Velocity.

observed dynamometer readings by this factor, and by plotting results with corresponding velocities, the curves shown in Fig. 225 are obtained. When P is the pressure in pounds per square foot, and V the velocity of the air-currents in miles per hour, the following equations representing the curves may be written:

For the Pitot gauge

$$P = .0025V^2, \quad \dots \dots \dots \quad (27)$$

for one model alone

$$P_1 = .0012V^2, \quad \dots \dots \dots \quad (28)$$

for the first model of a train

$$P_f = .001V^2, \qquad (29)$$

for the last model of a train

$$P_e = .00026V^2, \qquad (30)$$

for the second model of a train

$$P_s = .00008V^2, \qquad (31)$$

for any intermediate model between the second and the last of a train

$$P_i = .0001V^2. \qquad (32)$$

204. A Summary of Conclusions to be Drawn from the Work with Models. When a model having the proportions of a standard freight-car, or when a train of such models is submerged in currents of air, the length of the model or train being extended in the direction of the current, effects are observed which, briefly stated, are as follows:

1. The force with which the current will act upon each element of the train, or upon the train as a whole, increases as the square of velocity.

2. The effect upon a single model, standing alone, measured in terms of pressure per unit area of cross-section, is approximately .5 the pressure per unit area, as disclosed by the indications of the Pitot gauge.

3. The effect upon the different models composing a train varies with different positions in the train; it is most pronounced upon the first model; next in order of magnitude is its effect upon the last model; next, its effect upon each intermediate model other than the second; and last of all is its effect upon the second model.

4. The relative effect upon different portions of a train is approximately the same for all velocities. For example, any intermediate model other than the second always has a force to resist, which is, approximately, one-tenth that resisted by the first model, while the last model has a force to resist which is one-quarter that resisted by the first.

5. The ratio of the effect upon each of the several models composing a train, measured in pressure per unit area of cross-section, compared with the pressure per unit area disclosed by the indications

of the Pitot gauge, is, approximately, for the first model of the train, 0.4; for the last model of the train, 0.1; for any intermediate model between the second and last, 0.04; and for the second model, 0.032.

205. Atmospheric Resistance to Actual Trains.—Thus far attention has been directed to the effects produced by currents of air acting upon fixed models similar to freight-cars in outline and proportions, but much less in size. In what measure the results thus obtained will apply to trains of actual cars moving through still air is a matter yet to be considered. It is fair to presume that, had the models been larger than those which were really employed, the results observed would have been entirely consistent with those already given. If their dimensions had equaled those of a full-sized car even, there is no reason for supposing that the results obtained would have been disproportional to those which were actually observed from the smaller model, and it may be assumed, therefore, that the effects which would manifest themselves on a full-sized car of the same proportions with the model may be predicted with approximate accuracy from the known effects produced upon the model.

A full-sized car, having the same proportions with the models used in the experiments, would be a plain structure, 33 feet long and 9 feet wide, rising from a point close to the ground to a height of 12 feet along the center and 11 feet along the sides. When such cars are arranged in trains clear spaces of three feet would intervene between them. This combination of cars might be considered as representing for the present purpose an ideal train. The characteristics of an actual train, however, are difficult to define. Cars vary in the dimensions of their cross-section, in their length, and in the contour of their sides and roof. Box-cars are of simpler outline than coaches, and vestibuled trains present a more uniform cross-section than trains of platformed cars. Trains may be made up of cars of uniform size, or of cars each one of which may be so different in its proportions or outline as to produce an effect upon the atmosphere through which it moves measurably different from that produced by any other car of its train.

A careful review of the subject will show that differences in form or proportions existing between the model and the actual cars may not be greater than those existing between two different types of actual cars. The differences in effect arising from these differences in form and proportion, therefore, may be no greater in the former case than in the latter. If this is true the models will serve as a good

basis from which to make comparisons, and the belief is that the results which are given in succeeding paragraphs are not only sufficiently accurate for every practical purpose, but that they are as nearly true as any general statement applying to all conditions of service can be.*

Before proceeding to a consideration of details, it will be well to observe that estimates which have hitherto been calculated concerning the value of the resistance offered by the atmosphere to the progress of railway trains have been generally made upon a tonnage basis. An explanation for this is doubtless to be found in the lack of knowledge regarding atmospheric action. As the other resistances to which a train is subject are well expressed upon a tonnage basis, it has been convenient to express that which is of uncertain value in the terms of those facts which are better known. There is no justification for such a practice, for it is obvious that the atmospheric

* In comparing the conditions surrounding the model car with those existing about an actual moving car several points of difference are to be noted. First, the frictional resistance offered by the foundations of the model would tend to reduce the velocity of currents acting upon the lower portions of the model, in which case the force acting upon the model would be reduced below normal. Lower portions of an actual car, however, are required to pass through currents, which, because of their proximity to the ground, resist the motion of the car, so that the force acting upon such portions of the actual car are increased above normal. It would appear, therefore, that results obtained from the models would give values which are too low when applied to actual cars. Such a conclusion is undoubtedly justified, but the differences in question must be small, and they are probably neutralized by the fact that the body of the model car extends downward to the foundation, completely filling the space corresponding to that which, in the actual car, extends between the lower edge of the sills and the surface of the ground, giving enlarged side surfaces and enlarged area of cross-section.

Secondly, the presence of the foundation would, by frictional action, tend to reduce the velocity of all portions of the current in contact with or close about the models. Against this tendency is to be placed the fact that velocities were determined at a point where the flow occupied the full cross-section of the conduit. The presence of the models, with their foundations, reduced the effective area of the conduit, and necessarily gave a higher mean velocity to the current while passing them. How far the effects arising from these two facts will balance each other cannot be told, but that they are opposite is beyond question, and a study of conditions involved will show that neither can be very large.

Thirdly, the presence of the sides of the conduit may have served to constrain the wave-like action set up about the initial end of the train, and by so doing may have modified the effects observed for the last and for the intermediate models. Considering the relative cross section of the conduit and the models it is not likely that disturbances arising from this cause have led to any considerable error.

resistance for a loaded car is no greater than for a light car, values in either case depending entirely upon the size, proportions, and contour of the car.*

206. Application of Results Obtained from Models.—As the models experimented with were $\frac{1}{32}$ the size of a typical full-sized car, which, for the present purpose, may be assumed to represent any 33-foot box-car, the area of each surface presented in the actual car is $(32)^2=1024$ times the area of similar surfaces in the model. It is assumed that the effect of the wind upon solids of the same proportion will vary with the extent of exposed surface, so that the atmospheric resistance which will oppose the progress of the actual car will be to that which would oppose the progress of the model as 1024 is to 1. That is, if a is the force in pounds resisted by the model under the conditions of the experiments, and A the force due to atmospheric resistance to be overcome by the actual car under conditions of service, then

$$A=1024a. \qquad (33)$$

Expressions have already been written (equations 22 to 26) giving the force in pounds resisted by models under the influence of air-currents having a velocity of V miles an hour. Combining these with equation 33 gives the resistance in pounds, A, to be overcome by the actual car when moving in still air at a velocity of V miles an hour.

Thus equation 22, expressing the resistance offered by a single model, is

$$a_1=.000116V^2, \qquad (22)$$

and equation 33 gives

$$a_1=\frac{A_1}{1024}.$$

Therefore, for a single actual car alone,

$$A_1=.000116V^2\times1024,$$

or, approximately,

$$A_1=.119V^2. \qquad (34)$$

* There have been numerous attempts to express the atmospheric resistance of a railroad train in the form of an equation. None, so far as the writer is informed, have taken into account both the cross-section and the length of the train, or have attempted to distinguish between the head resistance and the frictional resistance of the intermediate cars. In most cases, also, the work has been based upon an assumed relation between pressure and velocity which has given values greatly in excess of those actually existing.

By a similar process there may be obtained:

For the first car of an actual train

$$A_f = .000097V^2 \times 1024,$$

or, approximately,

$$A_f = .099V^2; \quad \ldots \ldots \ldots \quad (35)$$

for the last car of a train

$$A_l = .000025V^2 \times 1024,$$

or, approximately,

$$A_l = .026V^2; \quad \ldots \ldots \ldots \quad (36)$$

for the second car of a train

$$A_s = .000008 \times 1024V^2,$$

or, approximately,

$$A_s = .008V^2;\text{*} \quad \ldots \ldots \ldots \quad (37)$$

for any intermediate car between the second and last

$$A_i = .000010 \times 1024V^2,$$

or, approximately,

$$A_i = .010V^2. \quad \ldots \ldots \ldots \quad (38)$$

It is to be observed that the constants appearing in the five equations immediately preceding are calculated to give directly the tractive force in pounds necessary to move typical cars, which are assumed to be the equivalent of any actual freight-cars, against the resistance of the atmosphere at any rate of speed, V being the rate of speed in miles per hour.

The atmospheric resistance for trains of such cars is the sum of the resistance of the several parts. Thus:

For two cars

$$A = A_f + A_l = 0.099V^2 + 0.026V^2 = 0.125V^2,$$

* This value is so nearly equal to that for A_i, that, in the work which follows, the resistance of the second car will be assumed to be equal that of any intermediate car.

for more than two cars, calling the number of intermediate cars b, or

$$b = \text{number of cars in train minus } 2,$$

$$A = A_f + A_i b + A_l = .099V^2 + .010V^2 b + .026V^2$$

$$= (.125 + .010b)V^2;$$

or, since the resistance of each intermediate car is a constant function of the velocity, the multiplier in the coefficient of V may readily be expressed in terms of the number of cars in the train. Thus, let

$$N = \text{number of cars in train},$$

then

$$b = N - 2,$$

and

$$A = (.125 + .010b)V^2 = (.105 + .010N)V^2;$$

the last form being, perhaps, somewhat more readily applied than the one preceding.

In a similar manner, by a suitable combination of equations 34 to 38, the atmospheric resistance of any train may be expressed.

207. Resistance Offered to Locomotive and Tender.—In the application of the equations given in the preceding paragraph, a locomotive and tender running alone may be regarded as two cars. In a train of freight-cars, headed by a locomotive and tender, the locomotive should be regarded as the first car and the tender as the second. Thus, the tractive force in pounds necessary to overcome the atmospheric resistance due to the motion of a locomotive and tender running alone is equivalent to

$$A = A_f + A_l = .099V^2 + .026V^2 = .125V^2,$$

which, for a speed of 40 miles an hour, gives

$$A = .125 \times 1600 = 200 \text{ pounds}.$$

The tractive force necessary to overcome the resistance of a locomotive and tender running *at the head of a train* is equivalent to

$$A = A_f + A_i = .099V^2 + .010V^2 = .109V^2,$$

which, at a speed of 40 miles an hour, gives

$$A = .109 \times 1600 = 174 \text{ pounds.}$$

208. Resistance Offered to Trains of Freight-cars.—A train composed of a locomotive, tender, and 20 freight-cars, would, in effect, be equal to 22 freight-car units. The resistance to be overcome would be that of the first unit plus that of 20 intermediate units plus that of the last unit. That is

$$\begin{aligned} b &= 20, \\ A &= A_f + A_i b + A_l \\ &= .099V^2 + .010 \times 20V^2 + .026V^2 \\ &= .325V^2, \end{aligned}$$

which, at a speed of 40 miles an hour, gives

$$A = .325 \times 1600 = 520 \text{ pounds.}$$

If it is required to find the force necessary to overcome the atmospheric resistance of only that portion of the train which is behind the tender, the resistance of the first unit (in this case the locomotive) and that of the second unit (in this case the tender) must be removed from the equation, that is, $A_f = 0$ and $b = 19$, so that the equation becomes

$$\begin{aligned} A &= 0 + 19A_i + A_l \\ &= .010 \times 19V^2 + .026V^2 = .216V^2. \end{aligned}$$

The resistance, therefore, opposing the progress of the 20 cars in a train, *following a locomotive and tender* at a speed of 40 miles an hour, is

$$A = .216 \times 1600 = 346 \text{ pounds.}$$

209. Resistance Offered to Trains of Passenger-cars.—The atmospheric resistance of a train of passenger coaches can be determined by reducing the number of coaches to an equivalent number of freight-cars. In general, it will be sufficiently accurate if each coach is made equal to two freight-cars. Thus, a train of five coaches following a locomotive and tender may be considered equivalent to 12 units, of which the locomotive and tender each count one. Numerical results may then be found as already described.

210. Resistance Offered to any Train in Terms of its Length.—It is evident that a car length of 33 feet as a unit of measurement is subject to some limitation. The equations already deduced, however, may be transformed into equivalent expressions, in which the length of the train is expressed in feet rather than in number of cars. Thus, considering the locomotive and tender as cars, the resistance of the whole train may be expressed in terms of the resistance of an intermediate car. If the actual number of cars is N, and the equivalent number of intermediate cars D,

D = number of intermediate cars equivalent to first car (locomotive) + number of intermediate cars + number of intermediate cars equivalent to last car

$$=\frac{.099}{.010}+(N-2)+\frac{.026}{.010}$$

$$=9.9+N-2+2.6=N+10.5.$$

So that the total resistance of any number of cars composing a train, when the locomotive and tender are each regarded as a car, is equivalent to the resistance of one intermediate car multiplied by the number of cars plus 10.5. But the resistance of an intermediate car is $.01V^2$, consequently that of the whole train is

$$A_{\text{(locomotive and train)}}=.01(N+10.5)V^2, \quad . \quad . \quad . \quad . \quad (39)$$

where A is the number of pounds tractive force necessary to keep the train in motion against the resistance of the atmosphere, N the number of 33-foot cars in the train, of which number the locomotive and tender are each counted one, and V is the velocity in miles an hour.

Again, if the length of the train in feet is represented by L, then

$$L=N\times 33,$$

or

$$N=\frac{1}{33}L.$$

Substituting this value of N in equation 39 gives

$$A_{\text{(locomotive and train)}}=.01\left(\frac{L}{33}+10.5\right)V^2=.0003(L+347)V^2, \quad (40)$$

which is the tractive force necessary to overcome the atmospheric resistance of the entire train when the length of the train in feet is known. Thus, a locomotive and train which measures 800 feet in length would be resisted, when running at a speed of 40 miles an hour, by a force of

$$A = .0003(800+347)1600 = 551.$$

By a similar process the resistance of the train following behind a locomotive may be expressed as

$$A \text{ (excluding locomotive and tender)} = .0003(l+53)V^2, \quad . \quad . \quad (41)$$

where l is the length in feet of the train, excluding locomotive and tender. Thus, if, in the example just assumed, the locomotive and tender were 66 feet long, the train following the tender would have been $800-66=734$. The pull of the tender draw-bar, when running at a speed of 40 miles an hour, would be

$$A = .0003(734+53)1600 = 378 \text{ pounds.}$$

In determining values for L and l, in equations 40 and 41, the length of the car bodies only is to be considered, since whatever resistance may arise because of the space between the cars is in each case included in the value of the constant appearing in the equation. It is unnecessary, also, to express the length of train with absolute exactness, since an error of one foot in the length of the train introduces an error of only one pound in the calculated result when the speed is 60 miles an hour; for a lower speed the error in the result arising from errors in the length of the train is less than this.

211. **Conclusions.**—The experiments already described, and the results deduced therefrom, justify certain conclusions. These, while stated in definite form, are in fact subject to a variety of conditions affecting their value, the significance of which is fully discussed in paragraph 205. It will be well to note in this connection that the conclusions here given apply to trains and parts of trains having an area of cross-section equal to that which is common in American practice; also that, being intended for general use, they should not be expected to apply strictly in any individual case. Their application may, in individual cases, lead to errors of from 15 to 20 per cent, but even with this limitation the conclusions given are vastly superior to any that have hitherto been offered; and, with this limitation also,

they will doubtless be found entirely sufficient for every requirement arising in practice. The conclusions are as follows:

1. The resistance offered by still air to the progress of a locomotive and tender running at the head of a train is approximately ten times greater than that which acts upon an intermediate car of the same train.

2. The resistance offered by still air to the progress of the last car of a train is approximately two and a half times greater than that which acts upon an intermediate car of the same train.

3. The resistance offered by still air to the progress of trains and parts of trains may be expressed in the form of equations, in which A is the tractive force in pounds necessary to overcome the resistance of the atmosphere, and V is the velocity in miles per hour. Such equations, in which the values of constants are given to two significant figures, are as follows:

(*a*) For a locomotive and tender running alone

$$A = .13V^2.$$

(*b*) For a locomotive and tender running at the head of a train

$$A = .11V^2.$$

(*c*) For the last car of a train of freight-cars

$$A = .026V^2.$$

(*d*) For the last car of a train of passenger-cars

$$A = .036V^2.$$

(*e*) For each intermediate freight-car in a train of 33-foot cars

$$A = .01V^2.$$

(*f*) For each intermediate passenger-car in a train of 66-foot cars:

$$A = .02V^2.$$

(*g*) For a train consisting of locomotive, tender, and freight-cars:

$$A = (.13 + .01C)V^2,$$

where C is the number of cars in the train.

(*h*) For a train consisting of locomotive, tender, and passenger-cars

$$A = (.13 + .02C)V^2,$$

where C is the number of cars in the train.

(*i*) For a train of freight-cars following a locomotive, but not including either locomotive or tender,

$$A = (.016 + .01C)V^2,$$

where C is the number of cars in the train.

(*j*) For a train of passenger-cars following a locomotive, but not including either locomotive or tender,

$$A = (.016 + .02C)V^2,$$

where C is the number of cars in the train.

(*k*) For a locomotive and any train, either freight or passenger,

$$A = .0003(L + 347)V^2,$$

where L is the length of the train in feet.

(*l*) For a train of cars, either passenger or freight, following a locomotive, but not including either locomotive or tender,

$$A = .0003(l + 53)V^2,$$

where l is the combined length of the cars composing the train.

4. A partial summary of results in convenient form is presented as Tables LXXXVII., LXXXVIII., and LXXXIX.

Table LXXXVII.

RESISTANCE OFFERED BY STILL AIR TO THE PROGRESS OF A LOCOMOTIVE AND TENDER.

Speed in Miles per Hour.	Locomotive and Tender Running Alone.		Locomotive and Tender Running at the Head of a Train.	
	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.
10	13	0.35	11	0.29
20	52	2.8	44	2.3
30	117	9.4	99	7.9
40	208	22	176	19
50	325	43	275	37
60	468	75	396	63
70	637	119	539	101
80	822	178	704	150
90	1050	253	891	214
100	1300	347	1100	293

TABLE LXXXVIII.

RESISTANCE OFFERED BY STILL AIR TO THE PROGRESS OF A TRAIN CONSISTING OF A LOCOMOTIVE, TENDER, AND CARS.

Speed in Miles per Hour.	Length of Train, including Locomotive and Tender.																	
	100 Feet		200 Feet.		300 Feet.		400 Feet.		600 Feet.		800 Feet.		1000 Feet.		1500 Feet.		2000 Feet.	
	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.
10	13	0.36	16	0.44	19	0.52	22	0.60	28	0.76	34	0.92	40	1.1	55	1.5	70	1.9
20	54	2.9	66	3.5	78	4.1	90	4.8	114	6.0	138	7.3	162	8.6	222	12	282	15
30	121	9.7	148	12	175	14	202	16	256	21	310	25	364	29	499	40	634	51
40	215	23	263	28	311	33	359	38	455	49	551	59	647	69	887	95	1130	120
50	335	45	410	55	485	66	560	76	710	95	860	115	1010	135	1390	185	1760	235
60	483	77	591	95	699	112	807	129	1020	164	1240	198	1460	233	2000	319	2540	406
70	657	123	804	150	951	177	1100	205	1390	260	1690	314	1980	368	2720	507	3450	646
80	858	183	1050	224	1240	265	1430	306	1820	388	2200	470	2590	552	3550	756	4510	961
90	1090	261	1330	319	1570	377	1820	436	2300	552	2790	669	3270	786	4490	1080	5700	1370
100	1340	358	1640	438	1940	518	2240	598	2840	758	3440	918	4040	1080	5540	1480	7040	1880

TABLE LXXXIX.

RESISTANCE OFFERED BY STILL AIR TO THE PROGRESS OF THAT PORTION OF A TRAIN WHICH FOLLOWS A LOCOMOTIVE TENDER.

Speed in Miles per Hour.	Length of Train, excluding Locomotive and Tender.																	
	100 Feet.		200 Feet.		300 Feet.		400 Feet.		600 Feet.		800 Feet.		1000 Feet.		1500 Feet.		2000 Feet.	
	Tractive Force.	Horse-power	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.	Tractive Force.	Horse-power.
10	4.6	0.12	7.6	0.20	11	0.28	14	0.36	20	0.52	26	0.68	32	0.84	47	1.2	62	1.6
20	18	0.98	30	1.6	42	2.3	54	2.9	78	4.2	102	5.5	126	6.7	186	9.9	246	13
30	41	3.3	68	5.5	95	7.6	122	9.8	176	14	230	18	284	23	419	34	554	44
40	73	7.8	121	13	169	18	217	23	313	33	409	44	505	54	745	79	985	105
50	115	15	190	25	265	35	340	45	490	65	640	85	790	105	1170	155	1540	205
60	165	26	273	44	381	61	489	78	705	113	921	147	1140	182	1680	268	2220	355
70	225	42	372	69	519	97	666	124	960	179	1250	234	1550	289	2280	426	3020	563
80	294	63	486	104	678	145	870	186	1250	268	1640	349	2020	431	2980	636	3940	841
90	372	89	615	148	858	206	1100	264	1590	380	2070	497	2560	615	3770	906	4990	1200
100	459	122	759	202	1060	282	1360	362	1960	522	2560	682	3160	842	4660	1240	6160	1640

CHAPTER XXIV.

A GENERALIZATION CONCERNING LOCOMOTIVE PERFORMANCE.

212. **Application of Data.**—From the data derived from Schenectady No. 1 it is possible to make estimates concerning the performance of locomotives in general, and to construct equations expressing performance with reference to several important functions.

213. **Boiler Performance.**—Dealing first with the matter of evaporative power, the following facts are to be noted: The records obtained from the Purdue locomotive disclose several tests for which the evaporation is above 12 pounds of water per foot of heating-surface per hour, and the maximum record for the boiler is 14.6. The results were obtained with Brazil block coal, which is of a light and rather friable character. While they were secured at the expense of very hard firing, it is probable that, with a superior grade of fuel, higher rates of evaporation could have been had, and that a possible maximum for this boiler need not be lower than 15 pounds. In accepting this limit it should be considered a possible maximum merely. It is at least three pounds higher than the practical maximum which can be relied upon in service for continuous work using Brazil coal. For short intervals of time the output of a locomotive boiler may be greatly increased beyond the normal maximum, but this is done at the sacrifice of fire condition or of water level. Power obtained by such means is the result of abnormal development, and is not a subject for consideration in connection with the present discussion.

From these considerations it is proposed to accept 12 pounds of water per foot of heating-surface per hour, as a fair measure of high performance for the Purdue locomotive under the ordinary conditions of the road, and as a close approach to the maximum evaporative power of all locomotives. With this understanding the value, 12 pounds of water per foot of heating-surface, will be accepted as a measure, by use of which the power of any locomotive may be predicted. It is

certainly one which can be accepted for all boilers of similar design. Probably, also, it will apply to all ordinary boilers in locomotive service, though this is not settled beyond doubt. It has been suggested that boilers having large grates may be easily forced to higher limits than those given, while boilers having a large extent of heating-surface and small grates may have difficulty in working up to the limit of 12 pounds. This argument is not without force, though probably it has less significance than would at first appear. Grate areas have an important influence on the efficiency of a boiler, but it is not clear that they can operate greatly to increase the power. If, in each of two boilers, one having a wide grate and the other a narrow grate, the same amount of heat is liberated, the heat in passing the tubes should in each case produce the same evaporation. On the other hand, the advantage of the large grate as a power producer appears in its ability to withstand forcing, and it must be admitted that an effort to secure maximum power in a locomotive is likely to resolve itself into a fuel-burning contest. In such a contest the larger grate is likely to have some advantage, but, for reasons stated, differences on this account will not be great.

On the other hand, very large locomotives, if hand-fired by a single man, can be made to work up to this limit of 12 pounds of water per foot of heating-surface per hour only with great difficulty and usually for intervals of time which are brief. In this case the limit upon the output of power is found not in the proportions of the locomotive, but in the strength and endurance of the fireman. The capacity of the locomotive is present, and may be utilized whenever means are found for feeding it. This inherent capacity is the factor for which a value is sought.

Finally, it should be said that the measure proposed cannot be a precise one. It is based on actual evaporation, which is necessarily influenced by changes in steam pressure and in temperature of feed, but it is evident that the requirements of the present process are not such as to make it necessary to take these into account.

From these considerations it will appear that the measure proposed is not likely to pass unchallenged; but, as one studies the problem, he will gain confidence in its value. For the present purpose, therefore, it will be assumed that the performance of all boilers may be predicted from the known extent of their heating-surface; that the measure proposed is not a measure of maximum performance, but is a close approximation thereto; and that the measure

is a fairly representative maximum for ordinary conditions of service. The measure expressed in the form of an equation is

$$\text{Water evaporated per hour} = 12 \text{ (feet of heating-surface in boiler), or}$$

$$E = 12H.$$

For example, a modern passenger locomotive having 3500 feet of heating-surface should, when working continuously at its maximum power, deliver per hour

$$E = 12 \times 3500 = 42{,}000 \text{ pounds of steam.}$$

214. Cylinder Performance.—Having now a measure of the weight of steam which a locomotive boiler may be depended upon to deliver, we may next make inquiry concerning the degree of economy attending the consumption of steam by the cylinders. The steam consumption per horse-power for the Purdue engine, for all speeds and cut-offs within its range of action, under a full throttle, has already been presented (Fig. 66, Chapter V). Assuming that this engine was designed to work at speeds varying from 25 to 55 miles an hour, and at loads necessitating a cut-off of from 6 to 10 inches, there are required from 26.28 to 32 pounds of steam per horse-power per hour. The maximum limit of 32 pounds is exceptional. A single value which fairly represents the actual performance of this 17×24 engine, working under 130 pounds steam pressure, is 28 pounds, and this value will for the present be accepted as representing the normal performance of a modern engine under its usual range of action. It is not the minimum value, for this is from 2 to 3½ pounds lower; nor is it the maximum, for the maximum may greatly exceed the measure stated. But it is, as has been stated, a value which represents the average consumption under such range of action as is normal to ordinary service.

We are now prepared to express cylinder power in terms of heating-surface. Thus, if 28 pounds of steam are required each hour for the development of a horse-power, and if each foot of heating-surface yields 12 pounds of steam per hour, then as many feet of heating-surface will be required for one horse-power as 28 will contain 12, or 2⅓. That is,

$$\begin{aligned} \text{Cylinder horse-power} &= \frac{\text{total pounds of water evaporated per hour}}{\text{pounds consumed per horse-power per hour}} \\ &= \frac{12 \times \text{square feet of heating-surface in boiler}}{28} \\ &= 0.43 \text{ (square feet of heating-surface).} \end{aligned}$$

Or, calling the cylinder power C.H.P., and the square feet of heating-surface in the boiler H, we have

$$\text{C.H.P.} = 0.43H. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

For example, the maximum cylinder power under continuous conditions of operation of a modern passenger locomotive having 3500 feet of heating-surface is

$$\text{C.H.P.} = 0.43 \times 3500 = 1500 \text{ (approximately).}$$

As this equation is one of great significance, it should be noted that the process leading up to its development has presented several steps, some of which have been subject to qualification. It is evident that the accuracy of the concluding statement cannot be greater than that of the parts of which it is composed. It does not apply with strict accuracy to any particular engine, or to any specific condition of running. It is merely an approximate measure of the maximum power which will be developed by a fairly representative modern engine under ordinary conditions of service. Its use is justifiable because of certain interesting and useful comparisons which it permits.

215. Draw-bar Pull.—When the horse-power and speed of a locomotive are known its draw-bar stress may be easily calculated, assuming no loss in transmission from the cylinder to the draw-bar. Thus, for any locomotive,

Work developed in the cylinders = the work given out at the draw-bar.

Therefore,

Cylinder horse-power × 33,000 × 60 = pounds stress in draw-bar × distance passed over in feet in one hour
= stress in draw-bar × speed of train in miles per hour × 5280.

Transposing,

$$\text{Pounds draw-bar stress} = \frac{\text{horse-power} \times 33{,}000 \times 60}{\text{speed of train in miles per hour} \times 5280}.$$

This is a general expression, true of any locomotive. Now, the horse-power of the typical engine has been shown to be C.H.P. =

$0.43\times$feet of heating-surface, and substituting this value for horse-power in the preceding equation,

$$\text{Pounds draw-bar stress} = \frac{0.43\times 33{,}000\times 60\times \text{feet of heating-surface}}{5280\times \text{speed of train in miles per hour}}$$

$$= 375\frac{\text{cylinder horse-power}}{\text{speed in miles per hour}}$$

$$= 375\frac{\text{C.H.P.}}{V}. \qquad (43)$$

This equation is assumed to be of general application. Applying it to the modern passenger locomotive employed in the previous illustration, having 3500 feet of heating-surface and developing 1500 horse-power, it appears that for a speed of 10 miles an hour such a locomotive will exert at the draw-bar a pull of

$$375\frac{1500}{10} = 56{,}250 \text{ pounds},$$

and at a speed of 50 miles an hour but

$$375\frac{1500}{50} = 11{,}250 \text{ pounds}.$$

The draw-bar pull for all speeds for such a locomotive is given by Fig. 226. This diagram, applying only to an engine of the dimensions stated, represents its maximum tractive effort, assuming that all the energy of the cylinders appears as a stress on the draw-bar. So far as the mathematical relations involved are concerned, the curve may be extended in either direction indefinitely, but practical considerations establish limits. The limit of its extent upward is found when the tractive force equals the adhesion of the locomotive, and the limit of its extent to the right is reached when the speed of revolution reaches a fixed maximum. Assuming the weight on drivers to be 96,000 pounds, and the adhesion to be one-fourth this amount, the maximum pull which the drivers will transmit is 23,750 pounds; that is, on the line *AB*, Fig. 226. The maximum speed is somewhat more difficult to fix. Assuming that the locomotive has 79-inch drivers, and that they may be allowed to travel as many miles per hour as they are inches in diameter, the limit of speed will obviously be 79 miles, cutting off the diagram along the line *CD*, Fig. 226.

Within limits thus defined the curve is a perfect definition of the theoretical maximum pulling-power of the engine under consideration. It shows that the engine may be started from rest at its maximum tractive force of 23,750 pounds, and may continue to exert this force until a speed of 24 miles an hour is reached. At this speed (point *B*) the power, which at lower speeds has been less than that which the

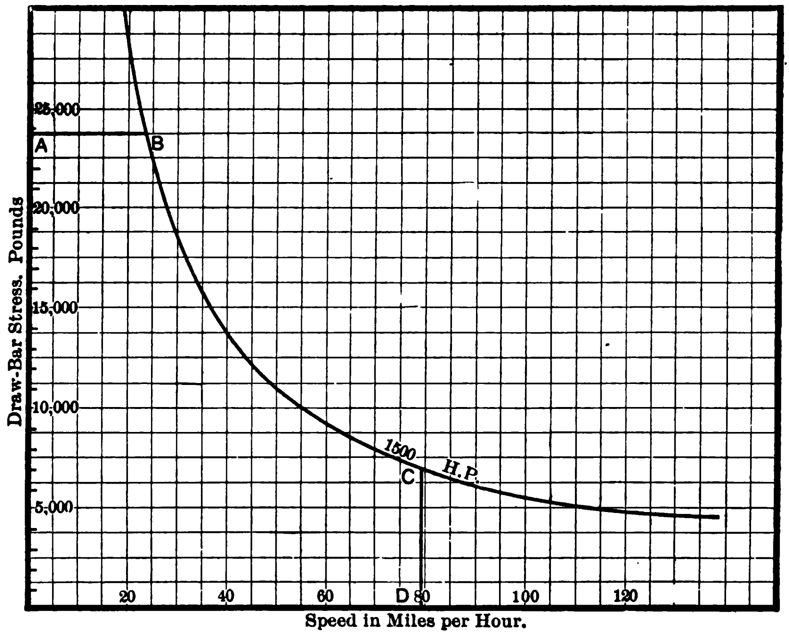

FIG. 226.—Draw-bar Pull as Affected by Speed.

cylinders are capable of exerting, becomes maximum at 1500 horse. Beyond this point each increment of speed is attended by a loss in tractive force, until at the maximum speed of 79 miles it becomes reduced to 7000 pounds. It is worthy of especial note that the reduction in draw-bar stress with increased speed is not due to any reduction in the power of the engine, but is in response to a physical condition which must always prevail; namely, that, when the available power is constant, the force in action must diminish as the velocity increases.*

* The proportions chosen for purpose of illustration in this and the preceding examples are those of a New York Central Atlantic type passenger locomotive.

216. Losses between Cylinder and Draw-bar.—Thus far in the discussion it has been assumed that no loss occurs in transmitting the power of the cylinder to the draw-bar. As a matter of fact, losses occur which are due to the following causes:

I. Machine friction of the engine, including the rolling resistance of drivers.

II. Rolling resistance of trucks and tender.

III. Resistance which the atmosphere offers to the head of the train.

There may be other resistances as, for example, the flange friction due to side-winds, which may result in loss of power, but these are exceptional and are constantly varying in value. For this reason they have no place in a consideration of the general case with which we are now concerned.

A general equation for machine friction may be written either in terms of mean effective pressure or of draw-bar stress. In anticipation of such an equation we may write

D = Diameter of drivers, feet;
d = " " cylinders, inches;
L = Length of stroke, feet;
N = Number of strokes in one revolution;
P = Mean effective pressure;
t = Draw-bar stress.

Remembering, also, that with no loss in transmission,

$$\left.\begin{array}{l}\text{The foot-pounds of work}\\\text{given out at the draw-bar}\\\text{in one revolution}\end{array}\right\} = \left\{\begin{array}{l}\text{The foot-pounds of work}\\\text{developed in the cylin-}\\\text{der in one revolution,}\end{array}\right.$$

it is evident that

$$t\pi D = PL\frac{\pi d^2}{4}N,$$

or

$$t = P\cdot\frac{d^2}{4}\cdot\frac{L}{D}\cdot N,$$

which will give the draw-bar stress whenever the mean effective pressure and the necessary dimensions of the locomotive are known. If, therefore, the value of P be made that which is just sufficient to keep the engine in motion against its own friction, its corresponding

value will be draw-bar stress equivalent to the machine friction. From conclusions elsewhere presented (Chapter XIX) it appears that for the Purdue locomotive Schenectady No. 1, the frictional resistance may be represented by a mean effective pressure of 3.8 pounds, and that this value remains fairly constant for all speeds and cut-offs. Whether the higher pressures of the more modern engine operate to increase this value cannot definitely be determined, but the indication is that the increase, if any, would be slight. In the absence of more definite information, we can do no better than to accept the value stated as of general application. If, therefore, the value 3.8 be substituted for P in the preceding equation, and if it is remembered that for simple locomotives N equals 4, the frictional resistance in pounds stress at the draw-bar may be written

$$t=3.8\frac{d^2L}{D}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

which as an approximate measure may be accepted as true of any simple locomotive.

For the Purdue locomotive the machine friction is

$$t_1=3.8\frac{289\times2}{5.25}=418 \text{ pounds},$$

while for a modern Atlantic type passenger-engine, previously employed as an illustration, having cylinders 21″ by 26″ and drivers 79″,

$$t_1=3.8\frac{441\times2.16}{6.6}=548 \text{ pounds}.$$

This force must be exerted at the draw-bar to keep in motion the pistons, cross-heads, valves, rods, and all connected machine parts, including the axles and flanges of coupled wheels. The significance of this statement is shown by the line CD (Fig. 227). It is independent of speed.

The resistance to rolling, offered by trucks and tender, cannot be much different from that of an equal weight of train following the tender. This is often expressed by the formula

$$t=2+1/6V, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

in which t is the resistance of each ton weight, and V is the velocity in miles per hour. This equation may be accepted as of general

application. To apply it the value of the rolling load must be known. The rolling load for the modern passenger-engine, hitherto employed as an illustration, exclusive of that carried by drivers, the resistance of which has been included with the machine friction, is as follows:

Weight on truck	42,000
" " trailing wheel	48,300
" " tender in working order	110,900
Total	201,200

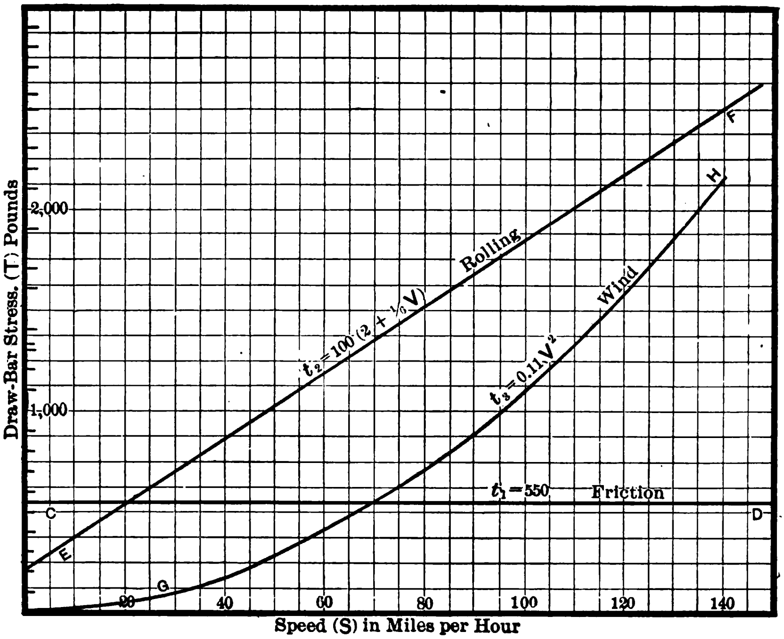

FIG. 227.—Losses between Cylinders and Draw-bar.

The formula shows that the tractive force necessary to overcome this rolling load of practically 100 tons, at a speed of ten miles an hour, is 366 pounds, and, for a speed of 50 miles an hour, 1033 pounds, values for all speeds being shown by the line *EF*, Fig. 227.

Experiments to determine the resistance offered by the atmosphere to the head of a train (Chapter XXIII) show that the progress of the locomotive and tender is resisted by a force approximately

ten times greater than that which acts upon an intermediate car of the same train, the formula being

$$t_3 = A = .11V^2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

where t_3 is the tractive force in pounds necessary to overcome the resistance offered by still air to the progress of a locomotive at the head of a train, and V the velocity of the train in miles per hour. The formula makes the resistance which the atmosphere offers to the motion of a locomotive, at a speed of 10 miles an hour, 11 pounds; and at a speed of 50 miles, 275 pounds, and for all speeds as shown by the curve *GH*, Fig. 227.

We have now obtained a general measure of the draw-bar stress equivalent to the power developed in the engine cylinders, assuming no loss in transmission, and have applied the same to modern passenger locomotives (Fig. 226); also values for the several losses which in service occur between the cylinder and the draw-bar (Fig. 227). If, now, we apply these latter measures as corrections to the former, the results should be a measure of the forces actually developed at the draw-bar. The effect of this process, as applied to the modern locomotive, is most easily made apparent by a graphical process, in which the curves of Fig. 227 are combined with that of Fig. 226. The result is shown by Fig. 228. In this figure the curve *AB* is the same as that given in Fig. 226. It is the draw-bar pull as calculated from cylinder work, disregarding engine friction and all other incidental losses. Applying to this curve the correction for machine friction which is defined by the curve *CD*, Fig. 227, we shall need to measure down on each ordinate a distance from the curve *AB* of 550 pounds. The result is the line *CD*, Fig. 228. Similarly, to make a further correction for the resistance of the rolling load, values from the curve *EF*, Fig. 227, are laid off downward from the curve *CD*, resulting in the line *EF*, Fig. 228. And, finally, to correct for wind resistance, values from the curve *GH*, Fig. 227, are laid off downward from the curve *EF*, resulting in the curve *GH*, Fig. 228. In the completed diagram the area between the first curve *AB* and the second *CD* represents frictional losses; that between the second *CD* and third *EF* the rolling resistance of trucks and tenders; that between the third *EF* and the fourth *GH* atmospheric resistance; and that between the fourth *GH* and the horizontal axes the forces which are available at the tender coupler for useful work in drawing a train. It will be seen that, at a speed of 25 miles an

hour, the stress at the draw-bar is 21,500 pounds; while, when the speed is increased to 75 miles an hour, the draw-bar stress is reduced to 5000 pounds.

Some insight as to the effect of proportions upon performance at speed may be had by comparing the percentage of the cylinder effort, which is lost when the speed is low, with that which is lost when the speed is high. Thus, at a speed of 25 miles an hour, the line *UV*

FIG. 228.—A Characteristic Diagram of Locomotive Performance.

is only 7.5 per cent of the line *UX*, while at a speed of 75 miles an hour the line *ZW* is 40 per cent of the line *ZY*. The relation of lost work to total work developed in the two cases is proportional to the length of the lines referred to.

217. The Application of Results to several Typical Locomotives.— The values shown by Figs. 227 and 228 are applicable to the New York Central Atlantic type passenger-engine only, but similar relationships may readily be developed which will be of general application. The general case can best be developed by means of the equations of the several curves involved, which are given together in

paragraph 218. The result of their application to four well-known engines is shown by Fig. 229.

A study of the diagrams of Fig. 229 will disclose some of the advantages which attend the use of heavy and powerful engines. Thus,

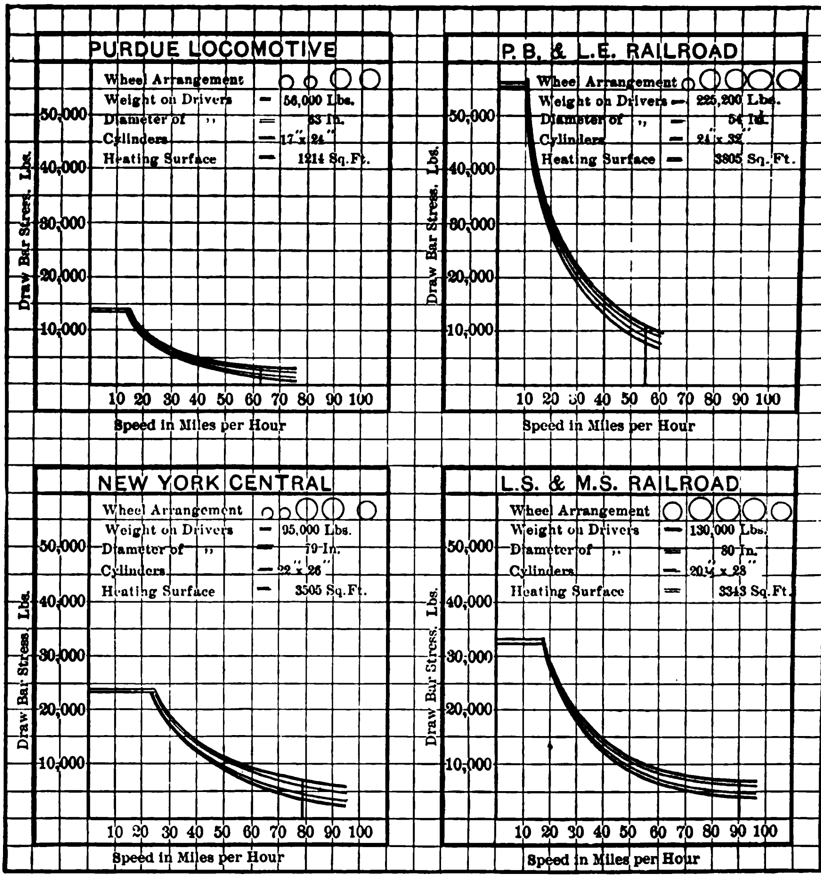

FIG. 229.—Characteristic Diagrams of Several Typical Locomotives.

comparing the performance of the Purdue locomotive, a representative engine of the year 1895, with that of the modern New York Central Atlantic type, it appears that, at a speed of 60 miles an hour, the actual draw-bar stress of the former is reduced to below 2000 pounds, while that of the latter is not less than 7000 pounds. At 70 miles an hour the pull of the Purdue engine drops to a thousand pounds,

whereas that of the New York Central engine is five times this amount. These are results due to the greater power available, and to the fact that the losses between cylinder and draw-bar are relatively less for heavier engines than for lighter ones.

Some of the effects of different diameters of drivers are to be seen by a comparison of the diagrams of the Pittsburgh, Bessemer & Lake Erie engine and the Lake Shore & Michigan Southern engine. The former, with its 54-inch drivers and its enormous adhesive weight, gives at slow speed a draw-bar pull of not less than 55,000 pounds, which is more than 50 per cent in excess of the maximum pull of the Lake Shore engine. The small-wheeled engine is able to develop the full power of its cylinders when a speed of 10 miles an hour is reached, whereas this result is not secured by the large-wheeled engine until a speed of 18 miles an hour is attained. Correcting for losses between the cylinder and draw-bar, it appears that, for a speed of 40 miles an hour, the tractive effort of the two engines becomes equal. Again, assuming the maximum speed in miles per hour to equal the diameter of the drivers in inches, the Pittsburgh, Bessemer & Lake Erie engine reaches its limit at 54 miles, whereas the Lake Shore engine continues to run until 80 miles are reached. Actual difference in respect to maximum speed would be greater than that which is shown; for, while the Lake Shore engine, with its large wheels and proportionately lighter reciprocating parts, may readily attain a speed approaching 400 revolutions, such a speed would be impossible with the smaller wheels and proportionally heavier reciprocating parts of the Pittsburgh, Bessemer & Lake Erie engine. The basis of comparison in this respect is, therefore, more favorable to the small-wheeled engine than it should be. If the maximum speed limit of the small-wheeled engine be fixed at 40 miles an hour, its entire range of action is represented by the area of the diagram to the left of the 40-mile line, which area is to be compared with the full area of the diagram of the large-wheeled engine. It will be seen that increasing the diameter of the drivers greatly increases the possible range of action, and, further, as it tends to diminish frictional losses, it serves in sustaining the draw-bar pull as the speed is increased.

218. **A Summary of Results,** expressed in the form of general equations applicable to any locomotive, is as follows:

Draw-bar stress based on cylinder performance,

$$\text{I.H.P.} = \text{C.H.P.} = 0.43H, \quad \ldots \ldots \quad (42$$

$$t = 375\frac{\text{I.H.P.}}{V}, \quad \ldots \ldots \ldots \quad (43)$$

$$t = 375\frac{0.43H}{V} = 161\frac{H}{V}. \quad \ldots \ldots \ldots \quad (47)$$

Corrections to draw-bar stress to be applied to cover loss in transmission from cylinder to draw-bar are as follows:

$$t_1 = 3.8\frac{d^2L}{D}, \quad \ldots \ldots \ldots \quad (44)$$

$$t_2 = W(2 + \tfrac{1}{8}V), \quad \ldots \ldots \ldots \quad (45)$$

$$t_3 = A = 0.11V^2. \quad \ldots \ldots \ldots \quad (46)$$

Draw-bar stress corrected for the several losses is as follows:

$$T_1 = t - t_1 = 161\frac{H}{V} - 3.8\frac{d^2L}{D}, \quad \ldots \ldots \ldots \quad (48)$$

$$T_2 = T_1 - t_2$$
$$= 161\frac{H}{V} - 3.8\frac{d^2L}{D} - W(2 + \tfrac{1}{8}V), \quad \ldots \ldots \quad (49)$$

$$T_3 = T_2 - t_3$$
$$= 161\frac{H}{V} - 3.8\frac{d^2L}{D} - W(2 + \tfrac{1}{8}S) - 0.11V^2. \quad \ldots \quad (50)$$

In these equations the characters employed signify as follows:

I.H.P. = indicated horse-power;
C.H.P. = cylinder horse-power, not necessarily obtained by means of an indicator, but estimated to be equivalent to power thus obtained;
H = square feet of heating-surface in boiler;
V = speed in miles per hour;
W = tons weight of rolling load, including that on trucks and tender, but excluding that on drivers;
w = tons weight carried by drivers;
d = diameter of piston in inches;
L = length of stroke in feet;
D = diameter of drivers in feet;
t = tractive force in pounds equivalent to the power developed in the cylinders, assuming no loss in transmission:

t_1 = tractive force in pounds equivalent to the machine friction, including the rolling friction of drivers;

t_2 = tractive force in pounds equivalent to the resistance of the rolling load, including that of trucks and tender, but excluding that of drivers;

t_3 = tractive force in pounds equivalent to the resistance offered by still air to the movement of an engine at the head of a train, being the same factor which is defined as A in section (*b*) of paragraph 211.

T_1 = tractive force in pounds equivalent to cylinder power corrected for machine friction. The application of the equation for t_1 to the New York Central engine gives the line *CD*, Fig. 228;

T_2 = tractive force in pounds equivalent to cylinder power corrected for machine friction and resistance of rolling load. The application of this equation for T_2 to the New York Central engine gives the line *EF*, Fig. 228;

T_3 = tractive force in pounds equivalent to cylinder power corrected for all losses between cylinders and draw-bar, including machine friction, resistance of rolling load, and atmospheric resistance. The application of the equation for T_3 to the New York Central engine gives the line *GH*, Fig. 228; T_3 must never be greater than $^1/_4 w$.

By solving for T_3, the net draw-bar stress for any locomotive, at any speed, may be found. The facts which are required to be known are, area of heating-surface, diameter and stroke of pistons, diameter of drivers, the weight of the rolling load, and the weight on drivers. No value for T_3 thus obtained should be considered practicable if greater than one-fourth the weight on drivers.

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ook may be kept
FOURTEEN DAYS
A fine of TWO CENTS will be charged
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